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THE ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

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SOLUTIONS TO SOME OF TO-DAY'S INDUSTRIAL PROBLEMS

15 *Industrial Problems are discussed in the* GENERAL ELECTRIC REVIEW, *November*, 1941, *Vol.* 44, *No.* 11. *The editor introduces this issue with the following words*:

"Many are the instances in which virtually the same engineering problems arise in widely different industries.

Knowledge of how the basic problem has been solved in a particular instance may well furnish a clue to how variations of it can be solved when encountered elsewhere."

In a preface L. A. UMANSKY says about the choice of the problems treated:

"The articles printed in this issue are on subjects taken at random from engineering files. They do not represent the most outstanding or the most spectacular achievements in their respective fields. On the contrary, an attempt was made to select typical, every-day problems, which industrial electrical engineers encounter. In each case the problem was solved a little more efficiently or by a somewhat simpler means—in brief, in a little better way than had been done previously."

The articles we are reprinting, in a condensed form, in this, and in the next issue of THE ENGINEER'S DIGEST, *were chosen in correspondence with the importance we attach to them from the point of view of our Readership.*

I.

Problem : How to pump power from a varying-speed source into a constant-frequency system.

Solution : Use a slip coupling between driver and generator.

Example : Application in economically loading airplane engines while being tested.

By G. E. CASSIDY, General Electric Company.

EACH airplane engine must be run under load for considerable periods. The only scheme now known for production loading of engines are calibrated propellers, water brakes, electric brakes and similar devices, all of which involve disposal of the engine power by complete wastage. Furthermore, engines are growing in size to the extent that the present power dissipating schemes of testing are beginning to present some difficult problems.

The following system of testing was developed in order to eliminate these difficulties. Since the first installation of this character consisting of two testing units, seventy-five additional units involving the same scheme of operation described in this article, have been or are in the process of being installed.

Solution of the Problem.

Ways of attacking the problem are: The engine might be connected to a d.c. machine and the machine connected to the power system through an adjustable-voltage drive. Well-known principles involving induction machines or synchronous machines with rotating stator as well as rotor might be resorted. But none of these possible drives compare favorably, from the standpoint of investment and simplicity with the drive illustrated in Fig. 1, which drive consists of a conventional synchronous machine tied directly to the power system, together with a slip coupling (a hydraulic coupling in the illustration) interposed between the engine and the synchronous machine.

In the operation of this equipment the synchronous machine may run as a motor or as a generator. With the coupling de-energized the synchronous machine is brought to speed and synchronized in the same manner as a conventional synchronous motor. Thereafter, the coupling is energized to make the synchronous machine "turn over" the engine. The engine is fired, and after it is warmed up it is brought to a speed above that of the synchronous machine. This machine, then acting as a generator, is loaded back on the power system by slipping of the coupling. The amount of this load is controlled by the engine throttle and the degree to which the coupling is energized.

To illustrate the distribution of engine power assume that the synchronous machine has a constant speed of 720 r.p.m., and that the largest engine to be loaded with the equipment (Fig. 2) is to be tested at speeds of 750 r.p.m., 900 r.p.m. and 975 r.p.m. Then, neglecting machine efficiency, which will not affect the general picture, the power distribution is as follows:

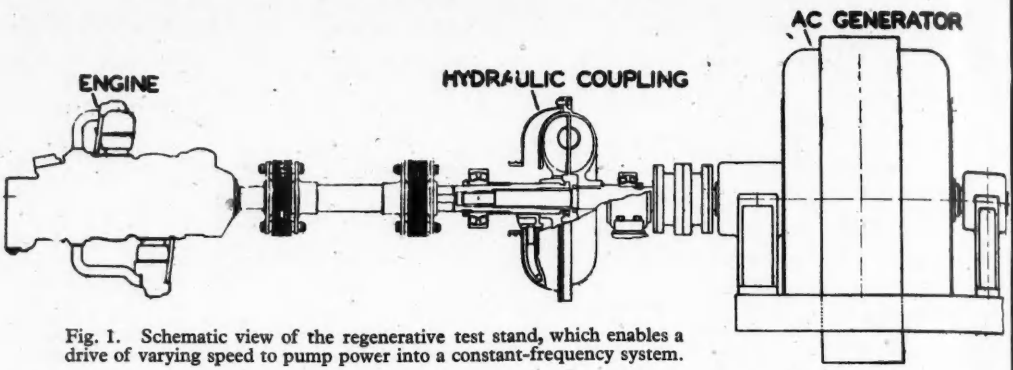


Fig. 1. Schematic view of the regenerative test stand, which enables a drive of varying speed to pump power into a constant-frequency system.

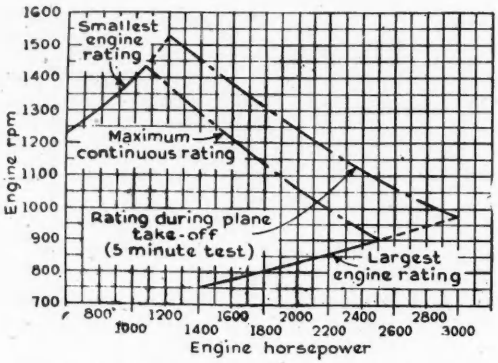


Fig. 2. Typical curves showing range of engine horsepowers and speeds for which testing equipment should be designed.

Engine Speed	Engine H.P.	Coupling H.P. loss	Generator H.P. delivered to power system
750	1400	75	1325
900	2500	500	2000
975	3000	780	2220

Thus at 3000 h.p. output of the engine all of the 3000 h.p. is not lost, as was the case with systems of the load dissipating type. Only 780 h.p. is lost.

If it is desired to load the engine below synchronous speed, this can be done up to the absorption capacity of the coupling by holding the rotor of the synchronous machine stationary—for example, by means of a brake. Fig. 3 illustrates a test run of a typical medium-sized airplane engine under such conditions, using a 900 r.p.m. generator. At low speeds the horse-power delivered by the engine is relatively low, so that the coupling will not be called upon to dissipate more horse-power than will be required under maximum running conditions. At low speeds, also, the duration of the test runs is short and horse-power hours lost in the coupling a relatively low proportion of the total loss.

The main object of the development of this new method of testing was economy. This was attained, and in addition a more effective way of testing large engines was also developed.

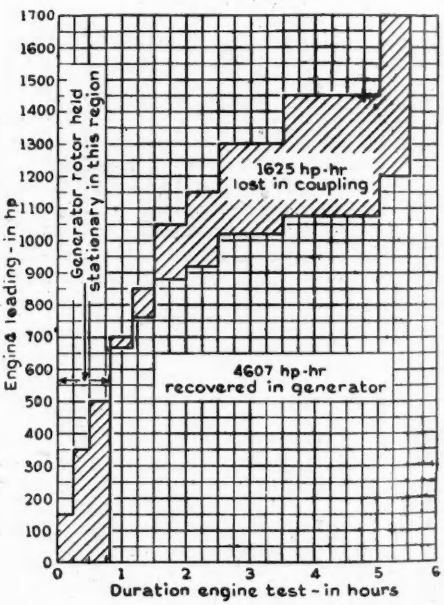

1625 hp-hr lost in coupling
Generator rotor held stationary in this region
4607 hp-hr recovered in generator
Engine loading - in hp
Duration engine test - in hours

Fig. 3. Graph showing that of the total of 6232 H.P.-hours delivered by an engine, during a test run, 4607 H.P.-hours were recovered in the generator.

In the scheme in Fig. 1 the hydraulic coupling was adopted for the first time in a new role. Previously, it had been used either at low slip values or on drives with rapidly decreasing power with increase in slip, such as fan drives. In the engine loading application it will be noted that power increases and increases rapidly, with increase in coupling slip.

II.

Problem : How to obtain an exceptionally compact and powerful adjustable-speed drive.
Solution : Use of water-cooled high-speed motors operating on high-frequency supplies.
Example : Application for drive of model propellers under test.

By C. C. CLYMER, General Electric Company.

FULL scale testing of new designs in actual flight is not only dangerous—it is also expensive. Fortunately problems connected with the development of new designs of planes, propellers or other parts can be solved, and variations checked in a wind tunnel, safely and at comparatively small expense.

The size of the test section of modern wind tunnel is limited principally by financial considerations. The power requirements of any two tunnels of equal efficiency vary as the square of the throat diameter, and as the cube of the wind velocity. Hence, with very few exceptions, planes and parts for testing in wind tunnels are built on a reduced scale, with rations as low as 1:15 being used.

These model planes are equipped with propellers driven by motors, just as they are on the full-scale counterpart, except that the model propeller is driven by an electric motor. The air speed on the model remains the same as for the full scale ship. Propellers for the model airplane are, of course, reduced in scale; but since the wind speed remains constant, the propeller tip speed must remain constant, with the result that the propeller speed must go up in the reverse ratio of the scale. The power, on the other hand, required to drive the propeller, will vary as the square of the scale. If, for example, the engine on a full scale plane were to have a rating of 2800 H.P., air-cooled, the engine nacelle might be approximately 50 in. in diameter. A 1/15 scale model would then have a nacelle diameter of about 3½ in. with a power requirement of 12.4 H.P., and speed, perhaps, in the neighborhood of 18.000 r.p.m.

The motor used to drive a propeller in such a model would therefore be approximately three inches in diameter, approximately 10 ins. long, and, since sufficient air would not pass through this small machine to cool it, water-cooling would be used. It would, perhaps, cease to resemble a motor, but this design has been found to do the job quite successfully. In fact, it could open up a new

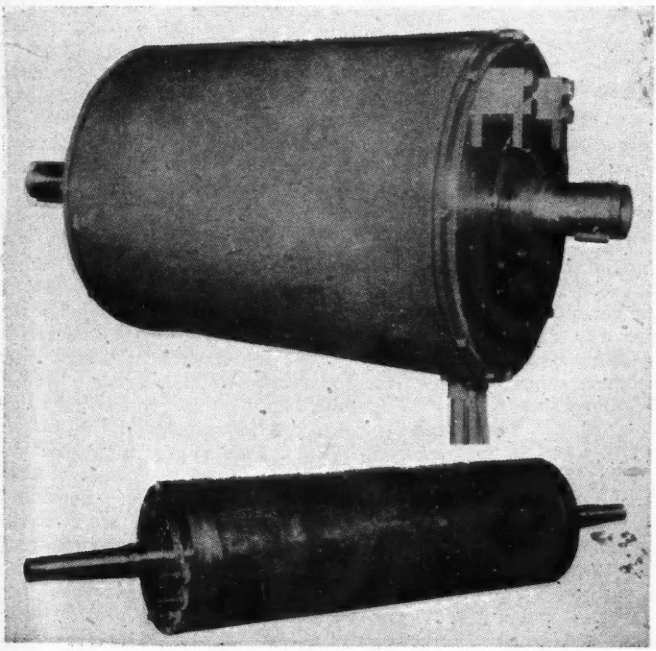


Fig. 1. Typical water-cooled motors used for driving model airplane propellers in wind-tunnel tests. The larger of the two motors shown is rated 1000 H.P., 2100 r.p.m., 105 cycles; the other 200 H.P., 5000 r.p.m., 83.3 cycles. Both are shown in about $\frac{1}{15}$th size.

field of motor application because of the small size and light weight.

The following ratings have been built and tested:

35	H.P.	4′	in diameter	10′	long	18.000	r.p.m.
130	„	8′	„	16′	„	5.400	„
200	„	10′	„	33′	„	5.000	„
1000	„	28′	„	38′	„	2.100	„

These motors are of the squirrel-cage induction type, and, as is apparent, are not designed for standard frequencies, but must operate from special high frequency power supplies. They can be 2-pole, 4-pole or 8-pole, depending upon the frequency of the supply. It is sometimes desirable to design the motors with a comparatively large number of poles in order that they will operate near the top frequency of the variable frequency supply, since this is the point where most power is available. The speed of the motor can be adjusted by use of an adjustable-frequency supply. As will be seen from the figures given above, the power requirements are lower for the motors with high speed ratings, which will be run from higher frequencies. This is unfortunate, since the power available from a given alternator will naturally increase as the frequency, and hence as the motor speed increases. This characteristic requires breaking up an energy source into a number of sets, the smaller sets operating at the higher frequencies. Hence, a low-frequency supply, ranging from 15 to say 120 cycles, would be chosen to provide perhaps 900 kW at 120 cycles 550 volt; for the frequency ranging from 120 to 400 cycles a generator of 200 kW at 400 cycles 550 volt may be suitable. These alternators, regardless of type, would be driven by a direct current motor taking power from an a.c. to d.c. motor-generator set, operating from a commercial power circuit. Adjustable-voltage (Ward-Leonard) control is usually provided.

Salient-pole alternators are used at frequencies around 120 to 150 cycles, while the inductor type is usually considered for the 400 cycle rating and above; converters have been used successfully at both 120 and 400 cycles, but above 400 cycles the inductor alternator seems to provide the best solution to the problem. Some saving can usually be obtained through the converter arrangement.

The proper procedure for starting is to reduce the frequency to the lowest convenient value and connect the motor directly to the line. After the motor has accelerated the frequency may be increased to the desired operating speed, without endangering the rotor by losses above the normal operating value.

III.

Problem: *How to improve tool performance.*
Solution: *Use of the electric Tool-pressure Gauge.*
Example: *Application to determine best Tool Shapes, Feeds, Depth of Cuts, Coolants, Machineability of Materials.*

By A. R. HAND, General Electric Company.

FORCE on the tool edge is directly related to rate of feed, depth of cut, surface speed of work or tool, shape and rake angle of tool, use of coolant or lubricant at the cutting surface, effect of dulling and heating of tool, and a knowledge of its magnitude is readily seen to be a most valuable asset in properly controlling conditions to obtain optimum operating performance and maximum overall efficiency.

By mounting the cutting tool bit in a holder at the end of a short stiff cantilever arm, and measuring the minute deflections of the end of this cantilever arm by electric gauges, a sturdy, dependable, and highly accurate pressure measuring gauge, the electric tool pressure gauge, could be obtained.

The instrument is shown in Fig. 1 and a schematic diagram of the tool holder and gauge elements in Fig. 2. The gauge consists of the following parts: (1) Gauge head, which comprises the tool-bit holder and gauge coils; (2) A balancing unit, which is a part of the electrical bridge circuit of the gauge; (3) An indicating and control unit which comprises the pressure indicating instruments and associated circuit controls.

The gauge indicates independently but simultaneously on two instruments the vertical and horizontal components of any force exerted on the point of the tool, whether these forces be steady or fluctuating. The gauge shown in the illustration is designed to handle any force up to 500 lbs. for either component, although gauges capable of handling much higher ranges of force can be readily constructed.

The principle of operation involves the unbalancing of a reactance bridge circuit by changes

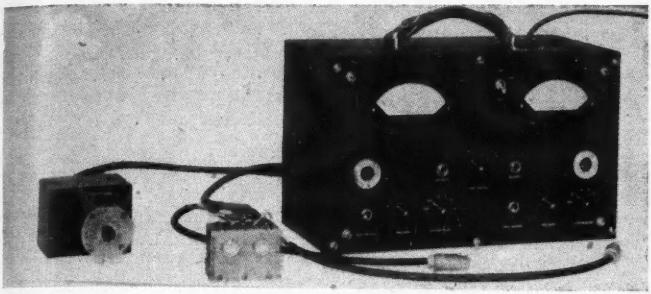


Fig. 1. The electric tool-pressure gauge, consisting of (left to right) the tool-holder, the balancing unit, and the indicating and control unit.

produced in the air-gap of one reactor arm under action of the pressure to be measured. The change in air-gap is made proportional to the change in pressure acting on the tool point, and the unbalance or output voltage of the bridge is fed to the pressure indicating instrument through a highly stabilized amplifier which insures accurate reproduction of the amplifier input in the output to the indicating unit.

Fig. 2 shows the mechanical system of translating tool pressure into air-gap variations, consisting of a short stiff cantilever arm which is anchored solidly at one end to the tool stock or compound. At the free end of the cantilever are attached a tool-bit clamp and the armatures of two small air-gap reactors. These armatures move a very small amount towards their corresponding reactors as the cantilever arm deflects in proportion to the pressures exerted on the tool point. The resulting change in air-gap alters the balance of the reactance bridges and causes the instruments to indicate the pressures. Two armatures and air-gaps are provided at 90 degrees to each other in a plane perpendicular to the axis of the cantilever arm. Each pressure component is measured entirely independent of the other. The only power supply required is an ordinary 115 volt 60 cycle outlet, since the frequency bridge circuit oscillator and the amplifiers are entirely self contained.

Applications.

The gauge has been found useful in the determination of the relative merits of various tool materials and tool designs, and in the establishment of operating practices which lead to optimum performance and increased output and efficiency. Typical examples of problems are the following:

(1) The determination of the most suitable form and design of tool for application to a particular kind of job.

It has been found, for instance, that the measured tool pressure is lowered as much as 25% by selecting the proper rake angle for the tool. This means that the power input to the machine and the heating of the tool and work are reduced very materially by the choice of proper tool design. It also has been shown, that the use of chip breakers of proper design reduces by as much as 20% the measured pressure on Carboloy tools.

(2) The determination of the relative merits of various tool materials, and differences between products of various manufacturers with respect to produce maximum material removal for a given power input to the machine, ability to stand up with the longest life, etc.

(3) The determination of the most favourable and efficient shop practices and methods for securing maximum output and maximum overall efficiencies.

(4) The choice of the optimum operating conditions with respect to selection of best rates of feeds, depth of cut, surface speed between work and tool, etc.

(5) The grading of various materials and alloys in terms of relative difficulty-of-machining index, corresponding to measured differences in tool pressure encountered in practice.

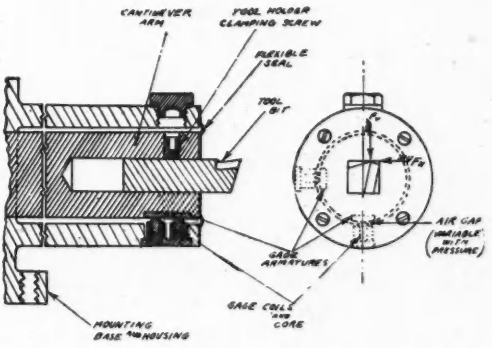


Fig. 2. Schematic diagram of the way in which the tool-holder and gauge elements are arranged.

IV.

Problem : How to obtain safer operation of large motors in hazardous atmospheres.
Solution : Use of inert-gas-filled motors.
Example : Motors in Oil Refineries and Gas-Gathering Plants.

By B. M. MILLS, General Electric Company.

SQUIRREL cage induction motors of explosion-proof construction to be located in proximity to inflammable volatile liquids or gases are available up to approximately 600 h.p. at higher speeds and meet the majority of the requirements for centrifugal pump and blower drives. Where larger induction motors are required or motors of other type, such as low-speed synchronous motors built integral with reciprocating compressors, a new design has been introduced recently using an inert gas, such as flue gas, nitrogen, or carbon-dioxide, in the motor enclosure under slight pressure. When combined with external cooling means, the size of the motor may not be larger than the equivalent open motor. The motor may therefore be built almost without limitation as to size and rating and without unduly increasing the cost.

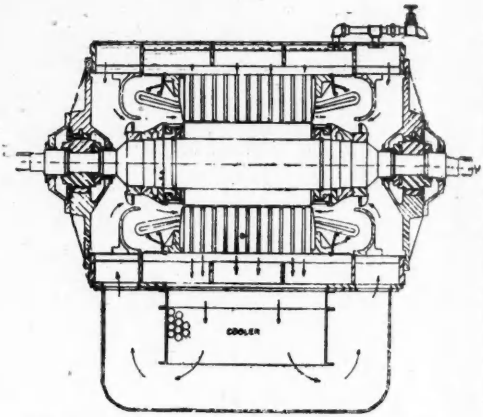


Fig. 2. Internal construction of an inert-gas-filled induction motor. Upper portion is a vertical cross section, the lower, a horizontal section showing cooling features.

Induction Motor Installation.

Two 1000 H.P. 3000 rpm carbon-dioxide-filled squirrel-cage induction motors, with built-in gas-to-water heat exchangers (Fig. 1) have been in successful operation in a California refinery for over two years' driving hot oil-pumps. Fig. 2 illustrates the design. The lower portion of the drawing is a horizontal cross-section on the centreline of the motor showing arrangement of the cooler tubes and direction of gas flow. The upper portion is a radial section showing valve at the top for purging air from the motor. Smaller motors may require only one heat exchanger, whereas the motor shown in the figure has heat exchangers on both sides. Propeller type fans maintain the circulation of the gas.

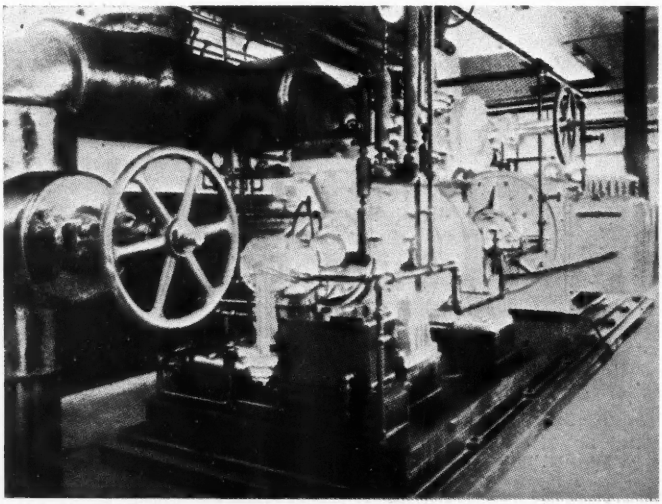


Fig. 1. Hot-oil charging pump in an oil refinery driven by a 100 h.p. 3,000 r.p.m. inert-gas-filled squirrel-cage induction motor with built-in heat exchangers.

Oil seals are provided on each bearing to minimize gas leakage. Separately motor-driven oil pumps —in duplicate for dependability— with an oil reservoir are provided. CO_2 is applied from a standard commercial high-pressure cylinder equipped with gauges, pressure regulator and reducing valves. Pres-

sure is reduced to a few inches of water. Motors are filled with inert gas by admitting the gas at the bottom of the motor and allowing air to exhaust from the purging valve at the top of the motor.

Synchronous Motor Installation

Three totally enclosed fan-cooled inert-gas-filled synchronous motors (two are shown in Fig. 3)each rated 100 H.P., 1.0 pf, 277 r.p.m. and built integral with reciprocating compressors, have been in successful operation for several years in a gas-gathering plant in Texas. The problem of enclosure was complicated by the necessity for a split end shield and seal, which could be assembled around the compressor shaft, yet retaining sufficient flexibility to allow for some shaft eccentricity. This problem was solved by the use of sectionalized, annular oil-resisting rubber diaphragms in combination with split carbon shaft seals, so that the leakage outward was limited to a few cubic feet per day. A design has recently been completed substituting the gas-to-water exchanger, as in the induction motor previously described, for the enclosed fan cooled method of ventilation.

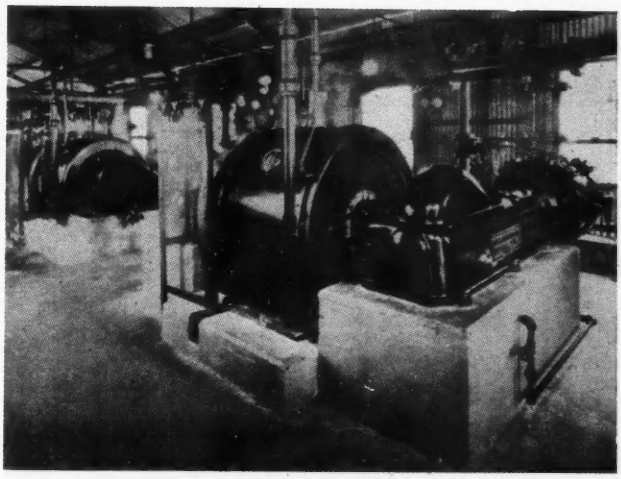


Fig. 3. Synchronous motors, inert-gas-filled, driving compressors in a gas-gathering plant.

V.

Problem : How to combine Copper Brazing and Heat Treating into a continuous Process.

Solution : Use a triple-chamber electric Furnace.

Example : Continuous Sequence of Brazing, Cooling, Reheating and Quenching of Sub-assemblies.

By H. M. WEBBER, General Electric Company.

SOME assemblies (Fig. 1) are made of high-carbon or carburized steels which must be heat-treated after copper-brazing in order to develop the desired physical properties. It is necessary to cool the steel from the copper-brazing temperature of about 2050 F to a point below the critical, say 1200 F, or lower, and reheat through the critical to refine the grain before quenching. To do this, it has been customary to braze and cool in one furnace and release for quenching in another. If these three operations could be performed in a single furnace, obviously handling costs would be reduced and production speeded up.

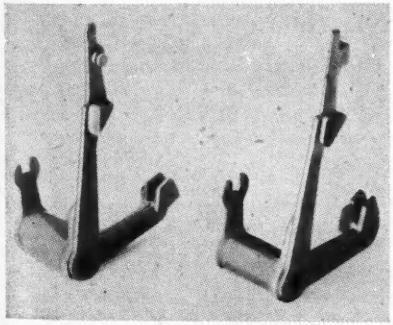


Fig. 1. Typical fabricated part copper-brazed and heat-treated by a continuous process in an electric furnace.

The Continuous Method.

The sub-assemblies generally have to be carried through the furnace on trays. Also, when reheated, they must be quenched directly from the proper heat treating temperature and not be allowed to partially cool before quenching. Therefore, it is necessary, in the continuous method, either to quench both the work and the trays after reheating, or to remove the trays and quench only the work, in a short interval of time. With small trays and a low rate of production box-type

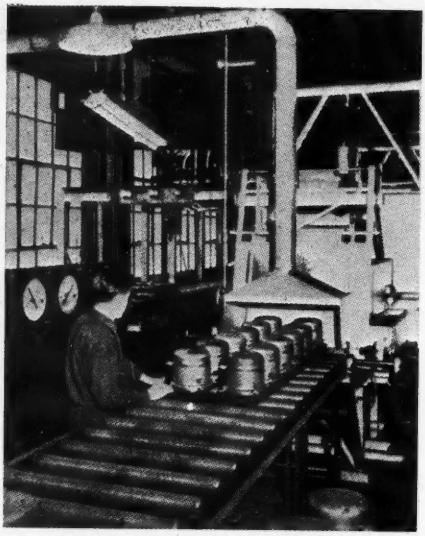


Fig. 2. Roller-hearth furnace with driven rolls, for continuous brazing, cooling, and reheating process.

furnaces are used, with the trays pushed manually through the brazing, cooling and reheating chambers. Then as the trayloads are pulled out the work can be quenched rapidly.

For high rates of production some sort of conveyorized furnace is desirable, in order to transfer the work automatically and minimize pushing and pulling stresses on the trays, and rails. For copper brazing the mesh-belt-conveyor type and the roller-hearth-conveyor type is in use. It would be logical to adapt one of these types by simply adding a reheating chamber to the brazing and cooling chambers. The roller-hearth furnace (Fig. 2) has many welcome qualities: low maintenance costs, high continuity of operation, long life of trays and rolls due to elimination of pushing and pulling stresses on the trays, and use of light-weight trays.

For these reasons, three triple-chamber roller-hearth furnaces were recently put in service in a large defense plant, each for continuously brazing, cooling and reheating fairly heavy sub-assemblies at a reasonable high rate of production.

Furnace Construction.

Each furnace (Fig. 3) has pneumatically operated end-doors, the opening being 22 × 6 in. The 6½ ft. long brazing chamber, operating at 2050 F, has rolled-ribbon alloy heating elements in the roof and upper side walls and on the lower side walls beneath the rolls. Its electrical rating is 85 kW divided into two zones in the length. Here the brazing metal, in the form of copper wire rings placed at the joints of the sub-assemblies, melts and creeps throughout the joints of the sub-assemblies by its ability to wet the steel in the presence of the reducing atmosphere in the furnace. Capillary attraction draws it through. The resulting iron-copper alloys which form in the bonds are very strong.

The 5 ft. long cooling chamber, in which the molten alloys solidify and cooling continues to about 1200 F, is lined with light-weight refractory brick and has alloy tubes on the walls through which air can be circulated to accelerate cooling.

The 7½ ft. reheating chamber has heating elements in the roof and lower side walls. Its electrical rating is 45 kW. A photo-electric relay signals the operator that a tray is ready to be removed for quenching.

Both heating chambers are equipped with automatic power and temperature control. Speed of the furnace roll-tables can be adjusted. The furnace atmosphere is supplied by an atmosphere-gas converter which burns an automatically proportioned rich mixture of natural gas and air. The resulting atmosphere is highly reducing and quite inexpensive.

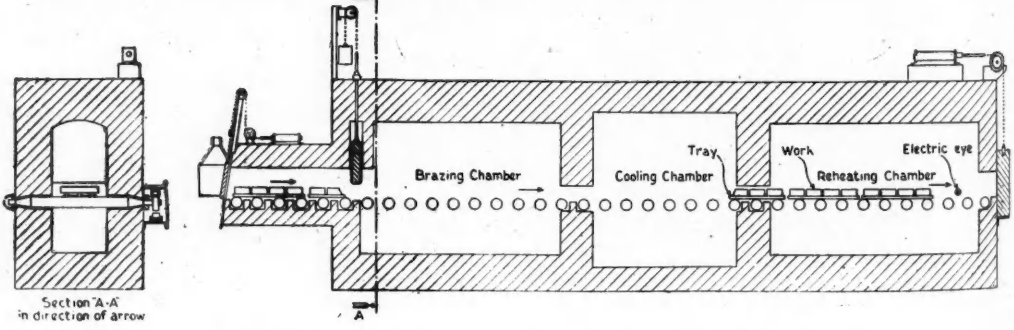


Fig. 3. Principal features of a continuous brazing and reheating furnace.

OF LIGHT METALS AND STEEL

By E. H. SCHULZ. (From *Stahl und Eisen*, Vol. 61, No. 50, December, 1941, pp. 1121-1125).

THE enormous progress made in aluminium production, reduction of manufacturing costs, development of aluminium alloys susceptible to heat treatment, and the persistent demand for a light construction material, especially in aircraft design, are jointly responsible for the vast application of aluminium by the industires. More or less the same could be said about Magnesium. The increasing utilization of aluminium and cheapening of its manufacturing costs in modern times is shown graphically in Fig. 1.

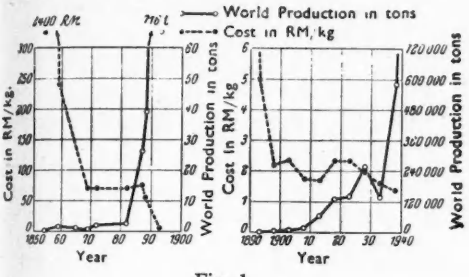


Fig. 1

It is only natural that such a remarkable progress should interest the layman too, and it is welcomed by the engineer that his work in this field is being brought to the attention of wide circles. Unfortunately, however, certain aspects brought forward in this fashion are misleading, though they appear correct at first inspection. So it is that the misconception has arisen of the "lightmetal age" replacing the "iron age." Another misleading statement frequently mentioned is that aluminium and its alloys have physical properties surpassing those of steel and, in particular, they have much superior corrosion resistance.

A very good example of a laymanish aspect of aluminium versus steel was exclaimed in an article on shipping (Z. Binnenshiffart, 1940), in which it was shown by means of calculation that construction of hulls from light metals would offer far going advantages, such as saving of useful tonnage, greater corrosion resistance, and even considerable saving of costs considering the better utilization of the ships.

The experts, of course, at once detected the fallacy of the statement, as the author of the article has compared pure aluminium with steel alloys when considering the question of economics, but strength calculations were strictly based on expensive aluminium alloys. The important part played by the strength modulus of aluminium was omitted altogether, and neither was any thought given that the cargo of ships is very often such material that would corrode aluminium much more readily than steel. Shortly, the arguments in favour of light metals for ship constructions could be overthrown at once by the expert.

So far the steel industry has failed to defend its own case, probably because practice goes different ways to theory, and hence no need for it, nevertheless, it seems pertinent to examine more carefully the relationship between light metals and steel, as they stand to-day. But it can be said already at this stage that there does not exist a competition between these materials; in reality, they follow parallel paths, and the demand of an application will decide from case to case whether one or the other material is the only one to give satisfactory results.

The following discussion is then based not only upon their physical and mechanical properties, but also upon an important practical consideration, namely upon the available quantities of raw metals. With regard to the latter, the crust of the earth contains about 7.3% aluminium, 5.1% iron, and about 2.5% magnesium. For manufacturing processes, however, the figures are not so favorable for aluminium, as here it must be remembered that the greatest portion of aluminium in the earth would not be practicable for extraction, the main source of aluminium production being bauxite, from which extraction can be made at reasonable costs.

Of still greater importance is the question of the extraction process. Without going into details, it can be stated that the energy expenditure for the production of 1 ton of steel or aluminium, expressed in tons of coal, is about 1.5 and 13 t. respectively At once this settles a considerable handicap on aluminium, being its price at about 8 times that of steel. Therefore, it is true to say that the prices of the raw materials are approaching a steady figure, especially as it is not to be expected that any unexpected discovery in the processing method might reduce the costs by a considerable amount.

The aluminium production in Germany in 1938 was about 1% of the total raw material pro-

duced. For the whole world this figure becomes still less favorable, and is not much affected if to this the magnesium production were added. At the same time it must be remembered that considerable quantities of raw material are required for numerous purposes, as, for instance, by the chemical industry for manufacturing paints, for alloying purposes, etc., and is thus further reducing the available supplies. Even if we assume that the world's production of light metals has been trebled since, it would be obvious that only a small fraction of the total steel requirement could be completely substituted by light metals, provided, of course, that their properties would permit this change. So here we have one of the most important arguments for steel.

But there still remains another side to the question how far can aluminium take the place of steel, where such a change would be desirable. This problem, of course, necessitates a thorough comparison of properties of the two materials involved, and at this point it should be emphasised that the statement that the strength of aluminium surpasses that of steel is completely false. Because there is a vast variety of steels, one cannot truly speak of strength of steel, and the mentioned statement must have been consequently arrived at by putting aluminium in opposition to mild steel (St. 37), a particular sort of steel. From a practical point of view this is most unsatisfactory, and the comparison should be made between best alloys of aluminium and steel. St. 37 could only be compared with pure aluminium, if at all.

Some comparative figures are given in Table I. For ease of comparison in the second column the respective cross sections of steel were made equal to 100%. It can be seen that in the case of hardened steel alloy the saving in weight by using aluminium is practically nil. Similarly it is remarkable that the tenacity of mild steel is more than four times that of aluminium (first column). It can be seen from this that where strength is a predominant factor, the use of steel is indispensable. Mild steel, on the other hand, could be replaced by steel alloys, thus resulting in saving of raw materials, but economic considerations often bar the way of doing so; to light metals this applies to a much greater extent.

Going a step further, from Table II. it appears that application of light metals in place of steel becomes even more undesirable. In Table II. St. 37 and St. 52 are compared with aluminium and aluminium alloys. Apart from the much lower notch impact strength and alternative bending strength of aluminium, its modulus of elasticity is also considerably less than that of steel, and, therefore, it cannot be used in applications where buckling might occur. In favour of light metals on the other hand the argument can be brought forward that they are readily machinable and are easy to cast.

Regarding the corrosion resistance of light metals, still much controversy exists at present. This property is namely largely influenced by the composition of the alloy, and, in general, it can be said that resistance to corrosion is much less of aluminium alloys than of the chemically pure aluminium. A few per cent. of iron and only a few hundreds of a per cent. of copper or nickel has an adverse effect on aluminium. There are, of course, some methods—surface treatments—that will improve the corrosion resistance of aluminium alloys, but at the same time one should remember that a large number of methods are also available for steel alloys with a similar effect. Certainly a comparison between specially treated aluminium and untreated steel would not be just.

Apart from economical considerations, there are a number of applications where steel only

TABLE I.

Material	Tenacity kg/mm²	Cross section at same strength %	Weight %	Yield Point kg/mm²	Elongation at Fracture %
St. 37	40	100	100	20	20 to 30
Technically pure Aluminium ..	9	440	160	3 to 4	20 to 30
Unalloyed hardened steel ..	60	100	100	33	> 20
Aluminium Alloy	28	214	77	18 to 20	15 to 20
Hardened steel Alloy ..	120	100	100	> 90	8 to 13
Aluminium Alloy	45	267	96	≤ 30	10 to 15

TABLE II.

	Steel		Light Metal	
	St. 37	St. 52	Aluminium	Aluminium Alloy
Notch Impact Strength mkg/cm² ..	15	10	9 to 10	2 to 5
Alternative Bending Strength kg/mm² ..	20	28	≤ 5	12 to 16
Modulus of Elasticity kg/mm² ..	≤ 20,000		6,000 to 7,200	

could be used as material, e.g., for tool steels, magnets, etc. Basically, steel can only be replaced by aluminium where high strength is not of great importance. As an example it might be used for tins ; but here again the question crops up whether sheet iron with a surface coating would not be a cheaper and better solution. Another field where a certain amount of competition may exist between steel and light metals is for castings. Strength considerations here distinctly favour aluminium, on the other hand the heavier weight of cast iron might be a deciding factor where this is desirable. So again we get to limitations of the use of light metals.

To complete this article a few words should be also said about the co-employment of aluminium and steel. As mentioned before, already aluminium is used as an alloying element in steel alloys. Although there are some difficulties encountered during casting and machining these alloys, nevertheless an increase of such applications can be expected in the future. A good example of co-employment of aluminium and steel is for overhead wires, where steel provided the strength and aluminium the electric conductivity.

The above mentioned points naturally reflect the technical point of view as it stands at present ; not much can be said about the future development. It is true that whereas the use of iron has a history of many hundred years and aluminium is a comparatively recent discovery as far as its utilization is concerned, the development of high quality steel alloys is also not much older.

It would be, therefore, more justifiable to speak of the " age of Metallurgy " in general than the " age " of steel or light metals. Modern advance is to explore and fully utilize various metals—a co-ordination rather than a competition between metals.

AUXILIARY HIGH PRESSURE OR INDEPENDENT TURBINE ?

By Dr. Ing. W. Weingartner. (From *Die Wärme*, Vol. 64, Nos. 51-52, December, 1941, pp. 471-474).

Whenever a decision is made to extend a steam power plant by means of increasing the steam pressure, the question arises as to whether to employ an auxiliary high pressure turbine or a separate turbine in conjunction with the existing turbines of the old power plant. In the following, some aspects will be discussed which should prove helpful in making the above decision.

1. Steam Pressure.

The auxiliary high pressure turbine utilises the heat drop which is available owing to the higher steam pressure. Reheating after the auxiliary turbine and before admission to the main turbine set is essential. The increase of power will, therefore, depend upon this additional heat drop ; its lower limit is fixed by the existing plant, the upper limit being variable within the scope that is possible considering the choice of maximum steam pressure.

The back pressure of the auxiliary turbine, in other words the inlet pressure to the main turbine, is of great importance regarding the steam consumption per additional unit of power generated (kg/kWh) as is shown in Fig. 1. It can be seen from this figure that by increasing the maximum steam pressure by 30 atm. the steam consumption per unit power generated is not reduced much.

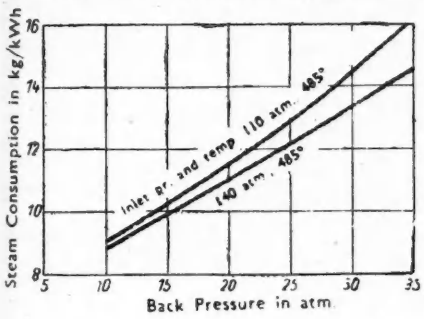


Fig. 1

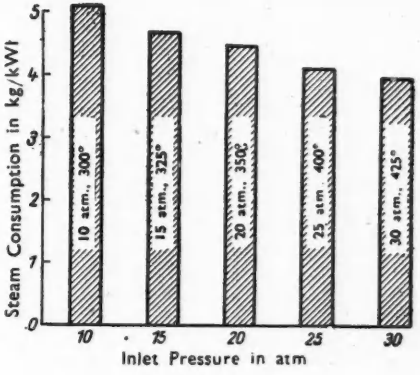


Fig. 2

From a practical point of view it is of greater interest to determine how much additional power can be generated by the new plant per unit power of the existing power plant. For this purpose Fig. 2 is drawn, showing the steam consumption of the main turbines. The pressure and temperature of steam indicated are taken from actual specifications, depending on how old the plant was. Then, from Figs. 1 and 2, the ratio of power from the old plant and new plant can be calculated. It must be remembered, however, that the total heat drop in the auxiliary plant is increased by heat loss during re-heating and transmission, and this must be allowed for when making the calculations. The results, on this basis, are given in Table I. and plotted in Fig. 3.

An inspection of the results clearly indicates a very important fact, namely, that with higher steam pressure to the main turbine the power increase rapidly decreases, that is the generation of power in the auxiliary turbine becomes more and more uneconomical. In any case the specific consumption of the main turbine is not so bad as to justify the high installation costs and additional operational difficulties arising from the auxiliary turbine.

The waste of steam by using an auxiliary turbine can be seen by considering that for the generation of 1 kWh the steam consumption is 10.7 to 16.2 kg when the back pressure is between 17 and 35 atm. respectively (15 to 30 at the main turbine), whereas the specific consumption of a separate turbine unit supplied with steam at 80 atm. is only about 4 kg/kWh. It also means that the boiler plant must be correspondingly bigger—at the same time working at higher pressure—to supply the auxiliary turbine.

TABLE I.

Inlet steam pr. at main turbine in atm.	10	15	20	25	30
Back pressure at auxiliary turbine in atm.	12	17	23	29	35
Steam consumption of auxiliary turbine at					
110 atm. in kg/kWh	9.5	10.7	12.3	14.1	16.2
140 atm. in kg/kWh	9.2	10.3	11.7	13.1	14.5
Steam consumption of independent turbine in kg/kWh	5.12	4.70	4.46	4.13	3.96
Gain in power per unit nominal power of old plant when inlet pressure of steam at auxiliary turbine is					
110 atm. in kW..	0.539	0.439	0.363	0.293	0.244
140 atm. in kW..	0.557	0.456	0.381	0.315	0.273

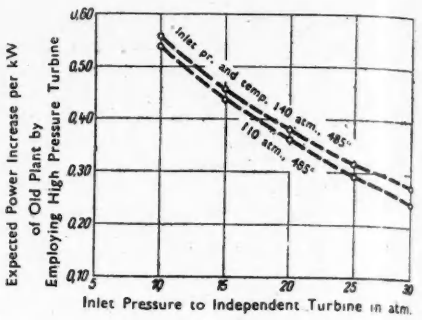


Fig. 3

2. Load. Thermal coupling of the old plant and the new auxiliary turbine is of little importance if the load is constant. In plants, however, where the load varies greatly, thermal coupling is undesirable. If namely one of the units thermally coupled has to be disconnected owing to the smaller load, the light load consumption might be fairly high, and therefore it would be more economical to use a separate condensing turbine without the necessity of re-heating. In this case, because the units are independent, the load regulation is much more economical as the light load consumption might be kept at a lower figure.

Fig. 4 shows a comparison between the old plant on one hand and the auxiliary or separate turbine on the other hand. The inlet pressure to the auxiliary turbine is 110 atm. Two turbines of 25,000 kW each are provided. If the total nominal power of the auxiliary turbine is utilized, then this gives 15,400 kW. This turbine has a light load consumption of approximately 63 t/h, i.e., more than 1/4 of the total consumption. The condensation turbine consists of two 32,000 kW turbines operating with a nominal steam pressure of 80 atm. To bring consumption figures to a comparable basis, the figures of heat consumption are given instead of steam consumption. From the figure it can be seen that only at loads exceeding 35,000 kW does the auxiliary turbine become more efficient than the independent turbine. But even at full load (64,000 and 65,400 kW respectively) the saving is only about 5%. About the same saving is obtained by the units compared with the old plant at 35,000 kW. Under 20,000 kW the old plant is actually the most efficient.

From this examination it becomes apparent that a comparison between the old and new plant on the basis of consumption would be misleading unless figures for the load variations are also given. Apart from the fact that the innovation of employing an auxiliary turbine gives no advantages

with variable loading, the use of independent turbines is a further step to simplicity and makes starting or stopping easier and quicker.

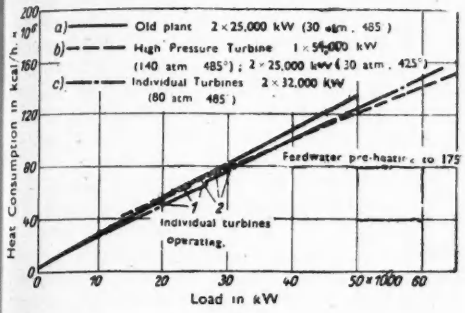

a) Old plant 2 × 25,000 kW (30 atm. 485°)
b) High Pressure Turbine 1 × 50,000 kW
(140 atm 485°) ; 2 × 25,000 kW (30 atm. 425°)
c) Individual Turbines 2 × 32,000 kW
(80 atm 485°)
Heat Consumption in kcal/h.
Feedwater pre-heating to 175°
Individual turbines operating.
Load in kW

Fig. 4

3. Room Economy. In some instances space may prove to be the deciding factor as to whether an auxiliary high pressure turbine or an independent turbine shall be adopted. In some cases it is not possible to increase the condenser capacity because of lack of cooling water or because there is no room for another condenser. Similarly, if there is no space for another turbine unit in the turbine hall, an auxiliary turbine must be used which can be fitted in the boiler room, especially as modern boilers are much smaller, power for power, compared with the older types.

4. Condition of Old Plant. So far it has been assumed that the turbines of the old plants are in sufficiently good condition to anticipate the satisfactory operation of these for an extended period when connected to the auxiliary turbine. But it must be remembered that it is not sound practice to attempt to rejuvenate the old and obsolete plant merely by the addition of an auxiliary turbine to it.

Regarding the boiler plant, if it is of insufficient capacity the employment of an auxiliary high pressure turbine is advantageous in that its exhaust passes to the main turbine. It can be seen, therefore, that the condition of old boiler plant contributes greatly to the economy of one or the other type of extension.

Opposed to this consideration, independent turbines permit full utilization of the old boilers, and if their output is not ample, the new boiler can be more generously designed. The two boilers then being connected through a pressure reducing valve the output of part of the new boiler will be available for the old turbines. Another important advantage of this system lies in the fact that the engineer has a free hand, not being tied by the back pressure which is fixed in the case of auxiliary high pressure turbines.

In some cases in addition to the auxiliary high pressure turbine an independent turbine is also provided to utilize the steam from the auxiliary turbine. But in this case one cannot truly speak of the turbine being an auxiliary. Such necessity might arise where the proposed extension is very large compared with the output of the old power plant. In such instances the efficiency of the auxiliary turbine becomes still less, and, therefore, such consideration alone would not justify this arrangement.

The sub-division of the heat drop in an auxiliary turbine and in the main turbine has one great advantage compared with the low pressure independent turbines, namely, that in case of a break-down, which is more likely to occur in the high pressure plants than in the low pressure units, the total output is not affected, provided of course that there is sufficient steam for the low pressure turbines.

Finally it can be said that auxiliary high pressure turbines are not advisable when the main turbines operate on moderately high pressures, not only because of the inefficiency as explained in the first part of this article, but also because new high pressure boilers have to be provided, thus putting out of commission some of the original boiler capacity.

DIESEL OPERATION AS VIEWED FROM VARIOUS ASPECTS

By A. PISCHINGER. (From *ATZ, Automobiltechnische Zeitschrift*, Vol. 44, No. 20, Oct., 1941, pp. 489-497).

IN the course of time various operation techniques have been developed for high speed diesels which are disclosing, in practice, certain advantages for one or another method. In the following the chief aspects for judging diesel operation will be surveyed.

Whilst a certain amount of consideration must be given to the problem of engine starting under unfavorable atmospheric conditions and to noise, the all important problem connected to diesel operation still remains the question of economy. The losses, affecting specific consumption, can be grouped as follows: mechanical friction losses, heat losses, such as exhaust gas losses, in-

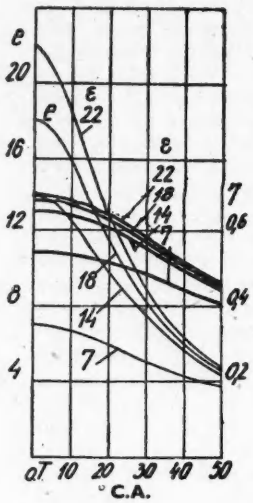


Fig. 1

complete combustion, radiation, etc., and pneumatic losses, as pumping losses, internal losses of flow.

Recent experimental work has proved that differences between the sudden pressure rises at the beginning of combustion are not responsible for increased frictional losses; but heat and pneumatic losses may greatly differ according to engine design.

The pressure in the cylinder at which combustion takes place is of great importance in respect of engine efficiency, and this is manifested in the aim of arranging high compression ratios. At the same time it must be remembered that combustion usually takes place after I. D. C., when the pressure is lower again. The effect of this can be seen from Fig. 1, showing the efficiency of an ideal engine for various values of C.r. (e) in dependance of retarding the combustion after the I. D. C. It can be seen that as an example if combustion is retarded 20° crank shaft angle after I. D. C. in effect this is identical with reducing the C.r. value from 22 to 14. Therefore, in practice, by increasing the compression ratio of an engine (within the practicable limits) will never bring about the same increase in efficiency as when care is taken to speed up combustion, and hence to bring it nearer to the I. D. C.

Combustion itself depends upon the quality of fuel and upon the efficiency of mixing the air-fuel charge. The latter provides, therefore, a means of controlling combustion, and the following factors affecting mixing should be considered:

(1) Fuel atomisation by means of the injection nozzle.

(2) Turbulence of the air charge, caused either by the compression stroke or by the locally varying intensity of combustion.

(3) Timing and location of fuel injection.

Although the creation of turbulence in the cylinder demands an expenditure of energy, the resulting speeding up of combustion more than compensates for it. Therefore, these two items (energy required for turbulence, gain by speeding up combustion) should not be separated on a balance sheet.

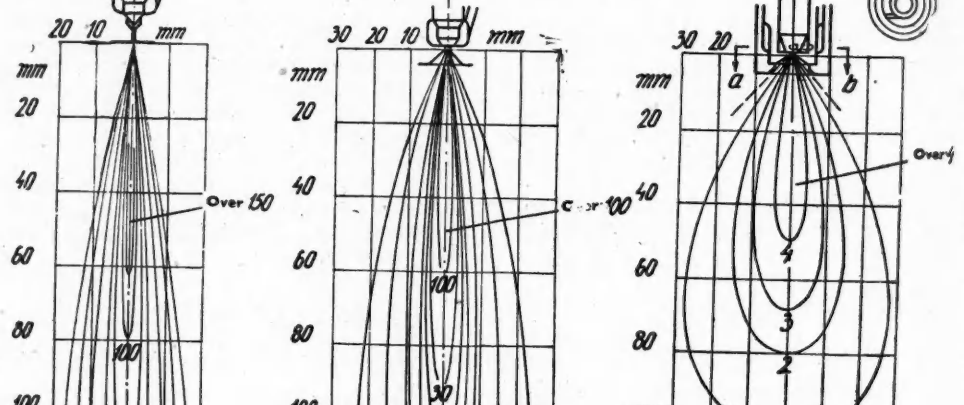


Fig. 2

Atomization of Fuel

The internal events causing evaporization and atomization are most complicated. Investigations so far carried out show that friction between fuel and air and the motion of fuel in the nozzle are both responsible for it. The latter factor becomes obvious, considering that atomization takes place also in vacuo. Another secondary factor promoting atomization is the sudden de-stressing of enclosed vapour particles and evaporation in the nozzle owing to increase of temperature.

The fuel distribution within the jet depends mainly upon the shape of the nozzle (Fig. 2). An elongated nozzle with a needle valve (Fig. 2a) gives a high fuel concentration at the centre of the jet. This can be reduced somewhat by increasing the rotary motion of the fuel in the annulus above the valve seat.

A short nozzle (Fig. 2b) will give a much less concentrated jet. The effects of shortening the nozzle is noticeable when the ratio of length/bore becomes 3. With further shortening of the nozzle the jet becomes unstable, that is, small irregularities might deviate the direction of the jet.

Fig. 2c shows yet another type of nozzle where the fuel is forced in a tangential direction into the nozzle. The effect is a great reduction of fuel concentration at the centre of the jet.

The penetration depth of the jet depends mainly upon the pressure into which injection takes place. Recent investigations have shown that with increasing back pressure the concentration of fuel at the jet centre increases first to a maximum, then gradually decreases. How far this phenomenon holds good is not known, but it was found that with some nozzle designs the fuel concentration increases up to the highest back pressures. In some cases the back pressure causes such changes in the fuel concentration that the jet characteristics are completely changed.

Combustion

The compression stroke causes air turbulence in all type of engines, and may be controlled or uncontrolled. By loosening the jet and hence further promoting mixing (employing multi-hole nozzles) uncontrolled turbulence was usually sufficient for combustion, but in modern high speed diesels this is no longer the case; combustion is delayed and incomplete (Fig. 3). Principally, two phases of combustion can be distinguished in Fig. 3. The first phase shows a sudden rise of pressure and in the second phase the speed of combustion rapidly decreases. If the ignition lag is reduced and injection continues into the combustion period, the result would be similar, with the alteration that the first pressure rise and the following combustion would be still further delayed. Although a sudden presure rise causes further turbulence and hence mixing, this advantage is counter-balanced by the high stresses caused in the driving mechanism, eventually leading to failure.

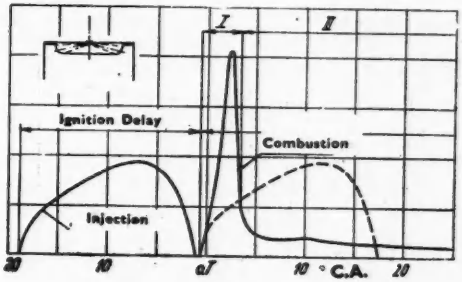


Fig. 3

The ideal case would be in contrast to Fig. 3 to shorten the ignition lag, increase the heat liberation/injected fuel in unit time and to increase the speed of combustion at the end of the process. However, the difficulty in solving this problem lies in the fact that it would demand measures carried out in a very short interval of time and bringing about effects opposing one another.

Methods of Reducing Shock Pressures

In order not to reduce the overall efficiency of the engine, one method only is available for reducing the first pressure shock—throttling the fuel during the ignition delay period. The amount of throttling will depend upon the details of the driving mechanism and on the noise. As a measure of pressure shock the audibility of the diesel knock is generally accepted, the knock being caused by the impact of the pressure wave upon the cylinder liner and piston. The audible diesel knock occurs when the rate of fuel injection is about 4-6 cm^3 per litre of stroke volume.

Endeavours to inject only a fraction of the total fuel charge during the delay period were unsuccessful, as it was not possible to get stable conditions with this arrangement. To-day injection systems operate with a reduced cross section of the nozzle for controlling the fuel quantity. A successful solution is the Bosch nozzle, shown in Fig. 4. When the needle is lifted from the seat an annulus is available for the

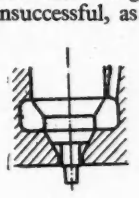


Fig. 4

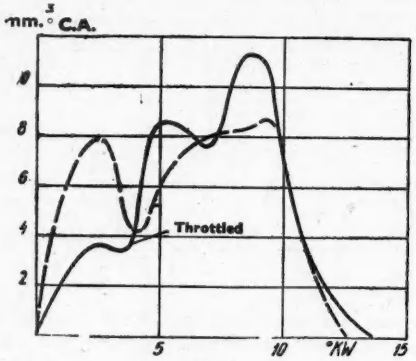


Fig. 5

fuel injection, and the full nozzle opening is effective only after the needle is lifted completely clear from the nozzle. Fig. 5 shows the quantity of fuel injected by the same pump when the nozzle is throttled and free.

Speeding-Up Combustion

Amongst the various types of diesel engines, the open chamber type (Fig. 6a) and the ante-chamber type (Fig. 6f) take up extreme positions. The possibility of using single hole nozzles and small injection pressures was achieved at the expense of higher fuel consumption and inferior starting of the engine. The ante-chamber design tried to eliminate these drawbacks. On the other hand improvement of the open chamber type, by improving the turbulence in the cylinder, has considerably increased the speed of such engines.

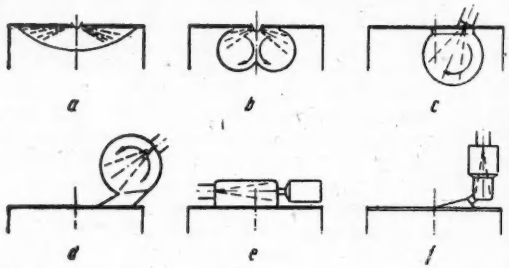


Fig. 6 : (a) original open chamber type ; (b) Saurer type ; (c) M.A.N. type ; (d) Older turbulence chamber type ; (e) Lanova type ; (f) Ante-chamber type.

The mixing of fuel and air charge in diesel engines is a combination of two ideal cases : the fuel is uniformly distributed in the combustion chamber by a suitable nozzle, or the fuel jet is directed to a point in the combustion chamber that is swept by a strong current of air, so that as a result uniform mixing of the charge occurs, similarly to mixing in an Otto engine. This method would not be practicable for diesels, as it would necessitate injection starting too soon. Therefore, the only possible solution is to arrange for a turbulence which, together with the suitable jet, will complete thorough mixing during the short interval of time that is available for this. The importance is, therefore, divided between a suitably arranged jet and the turbulence.

From this it follows that a co-ordination between the jet and air turbulence is of great importance, and the following should be noted: the weaker the air current, the better must be the distribution of fuel in the jet. If the air turbulence varies locally within the combustion chamber the core of the jet should be directed towards the point of greatest air velocity. The length of jet should be adjusted, taking into account the diminution of penetration power of the fuel droplets by evaporation during their path from nozzle to cylinder wall.

The beginning and period of injection should be timed to suit the air currents, that is, rapid and thorough mixing should be finished before combustion starts ; this applies especially to slow and moderate speed engines.

Open Chamber Type

The turbulence caused by the compression stroke will depend upon the amount of contraction of the cross section from the cylinder bore to the cavity in the piston head forming the combustion chamber. At the same time the shape of the cavity is of great importance, and should be in accordance with the air flow ; as an example, with an eccentrically located cavity on the piston head, the best shape is spherical for maximum turbulence. The jet should be so directed as to penetrate the region of fastest air current, and should start when the highest air speed occurs. Both the beginning and cut-off of injection are sharply defined, and injection should finish shortly after beginning of combustion. The completion of mixing is the result of the sudden pressure rise after ignition and, therefore, with this type of engine a damping out of the first pressure shock is not desirable. Such engines will be rough running if good efficiency of combustion is required.

Ante-Chamber Designs

The very narrow throat between main chamber and ante-chamber results in high speed air currents to and fro the ante-chamber. As opposed to the open chamber type the ante-chamber encloses only a fraction of the total air

charge, and therefore mixing in and outside the ante-chamber are important factors. Injection starts while air is forced into the ante-chamber, and continues over a period extending well into the combustion phase, the latter starting near the I. D. C.

Ignition is preferably initiated near the nozzle, and the volumetric change produced, rapidly overcomes the pressure difference between main and ante-chamber, thereby providing the blast necessary to drive the combustible mixture into the main chamber.

For efficient mixing in the main chamber it is essential to arrange next to a well guided air flow, that a great portion of fuel be concentrated just in front of the ante-chamber throat. The inclination and diameter of throat, the length of jet, the shape and timing of jet, are all important factors which must be brought into co-ordination. For such engines loosened jets are not suitable, as these would not be able to carry sufficient fuel to the throat entrance.

The importance of the first pressure shock is much less here than in open chamber engines, and this type of engine is, therefore, especially suited for smooth, noiseless operation.

Turbulence Chamber Engines

Here the cross-section of the throat is enlarged, and mixing is performed almost completely in the auxiliary chamber into which practically the whole air is compressed. This design takes up an intermediate position between the open chamber engine and the ante-chamber type; the air velocity is only about half that of the ante-chamber type, the injection extends well into the combustion period. The jet is directed to the entrance of the throat, but these engines are not much influenced by the fuel concentration in the jet.

As mixing is almost complete in the auxiliary chamber, the blast into the main chamber is of secondary importance as far as turbulence is concerned. Specially designed piston heads with cavities were without effect, except that this increased the air volume remaining in the main chamber and, therefore, the design became more like an open chamber type.

Fuel Consumption, Power

The maximum velocity of swirl occurring with the open chamber, turbulence chamber and ante-chamber type are about 150, 250 and 450 m/s. respectively. The kinetic energy of air flow is consequently increasing up to the I. D. C. (maximum air velocity), and amounts to about 0.05, 0.15-0.2, and 0.3-0.4 kg/cm^2 per unit piston surface area for the above three types of engines. During the return stroke of the piston the gases are once more set in motion and the energy expenditure is about the same as during the compression stroke. Only a small fraction of the kinetic energy is recovered in the form of mechanical energy, and should, therefore, be regarded as complete loss. At the same time an increased heat loss through the cylinder walls to the cooling liquid presents itself and, therefore, these two kinds of losses are difficult to separate.

Naturally the fuel consumption will be correspondingly higher by the amount of the energy losses required for thorough mixing of the fuel-air charge; the fuel consumption amounts to about 190-200, 170-190, and 160-180 g/H.P. hr. for the three types of engines respectively. However, the losses inherent to mixing are more than compensated, not only by the resulting higher efficiency of operation, but also by the fact that with good mixing, less excess air is required for combustion, so that an overall gain in power output is obtained.

Operation at Various Speeds.

It is impossible to arrange equally good working conditions over the whole speed range in regard to consumption and maximum load; usually the engine is so regulated that the minimum fuel consumption is obtained at the speed at which the engine is most frequently running (about 2/3 full speed). If the speed is higher than this the fuel consumption increases and power decreases. At lower speed than this care must be taken to supply sufficient air charge for combustion, and, therefore, the inlet valve is adjusted to close earlier as would be desirable when running at top speed. The fuel charge on the other hand is so regulated that the turning moment increases with speed, a desirable feature not only because this eliminates frequent changing of the gear, but it improves the acceleration of the engine.

Timing of injection: this is a tricky problem, as hand regulation would make efficient operation dependant upon the skill of the driver, and automatic regulation would deteriorate with time owing to wear. Great improvement is achieved to this end with the ante-chamber and turbulence chamber engines, as these are not sensitive towards beginning of injection, which can be adjusted to a constant value throughout the entire speed range.

The timing of beginning of pumping stroke, injection and combustion vary not only with design but within the same engine with speed.

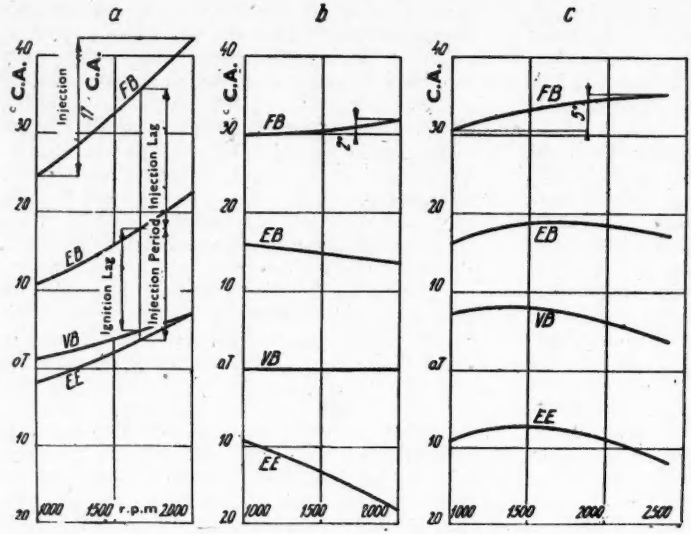


Fig. 7 : (a) Open chamber engine ; (b) Ante-chamber engine ; (c) Turbulence chamber engine. FB = pumping starts ; EB = injection starts ; VB = combustion starts ; EE = end of injection.

Variation of the timing of these factors is shown in Fig. 7, from which it becomes apparent that the ante-chamber and turbulence chamber engines require much less adjustment with speed than the open chamber type.

With all types of engines, employing auxiliary combustion chambers, the air current into the auxiliary chamber continues until I. D. C. is reached, especially if combustion does not start before this. With increasing speed the mass of air forced into the auxiliary chamber is reduced and, in consequence, the energy blast is reduced during the combustion stroke. In addition, owing to the increased pressure difference between main and auxiliary chamber, the blast and consequently the combustion is delayed adding to losses of power and efficiency.

Such limits of speed exist for all types of engines, ncluding the open chamber type if the contraction to the cavity in the piston head is considerable. This limitation of speed is more severe, the longer the path of air flow to cover all the air available for combustion. Small engines with compact combustion chamber designs can be run, therefore, at higher speeds with good efficiency figures than larger units. From an efficiency point of view, the ante-chamber type appears to inhibit for this reason speeds higher than usually adopted to-day.

An appreciable increase of frictional losses at higher speeds is not to be expected on the basis of present experiences.

Cold Starting.

Considering starting of engines under unfavorable atmospheric conditions, two phases should be distinguished and examined from a different angle ; the ignition of the charge, and the actual starting of engine (picking-up).

The temperature at the end of the compression stroke, which is to be regarded as a minimum for ignition to take place, varies with engine type and speed.

The temperature reached is influenced by the atmospheric conditions (temperature of air), the compression ratio and the heat losses to the coolant through the walls. To reduce the heat losses to the coolant the wall surface area should be low compared with the enclosed charge.

In general the heat losses during starting in a direct injection type of engine are lower than in an ante-chamber type. The turbulence chamber type, in this case too, takes up an intermediate position between the two other engine types. The lowest atmospheric temperature at which ignition is possible without auxiliary arrangements is about +15° C. for the ante-chamber type, 0° C. for the turbulence chamber type, and about -10° C. for the open chamber type. Alterations of these temperatures are subject to speed, which is usually between 60 and 150 r.p.m. during starting. So far it has not been possible to design diesel engines which could be positively started without hot bulb or other auxiliary arrangements at all practical temperatures.

Once ignition is started the engine will start running if the energy of combustion liberated is sufficient to overcome the internal resistance of the mechanism which is mainly the viscosity of oil. This resistance, too, will depend largely upon the surrounding temperature and might be, in some instances, much higher than the available turning moment of the engine during

starting. While the engine is still cold, mixing and combustion are uncontrolled and incomplete, and from this point of view there is not much difference between the various types of engines. In general, engines with higher consumption and power output are more easy to start, and in smaller engines it is advisable to inject an additional fuel charge during starting.

RUBBER IN MACHINE CONSTRUCTION

By DIPL. ING. G. EHLERS. (From *Kunststoffe*, Vol. 31, No. 12, 1941, pp. 422-424).

THE nature of the application usually determines certain specific requirements of the material to be used, such as, for instance, mechanical strength, chemical composition, temperature resistance, etc. When the working material is rubber all these requirements should be more carefully studied beforehand, as generally only a particular sort of rubber will answer the requirements satisfactorily. The properties of rubber being primarily a problem of processing, it is obvious that a close collaboration should be maintained between the designing engineer and the rubber manufacturer. The rubber processing will then be such as to bring about the required improvement of one particular property, remembering, however, that this is usually linked with a deterioration of another property of the material.

In the following some general properties of rubber will be discussed, to which consideration should be given before employing rubber machine parts.

There is a comparatively narrow temperature range only within which the application of rubber is permissible. Exceeding the upper limit is much more dangerous than exceeding the lower one. According to the composition of rubber it freezes at -20 to -70° C., and becomes then hard and brittle; but on thawing, rubber regains its original properties. Not so at high temperatures; at about 200° C. rubber becomes pasty and glutinous, Buna rubber becomes hard, but, after cooling down, these properties are still retained, and the material is for all practical purposes destroyed.

When rubber is exposed to some fluids, e.g., petrol, it swells, and the accompanying change of weight and volume will adversely affect the mechanical properties of rubber, leading eventually to complete destruction of the material. The effect is aggravated at higher temperatures. It is advisable, therefore, to keep the surface in contact with the attacking liquid as small as possible. There are grades of rubber which are resistant to the chemical action of petrol and other substances, as, for instance, Perbunan rubber, but the tensile strength and elasticity of these are inferior to those of natural rubber.

Special attention should be given to ageing of rubber. Whilst natural rubber under the action of heat shows a tendency to soften, synthetic rubber hardens. As can be seen from Figs. 1 and 2 ageing at high temperature leads to the destruction of the mechanical properties of rubber, the effect being much more marked with natural rubber as compared with synthetic rubber.

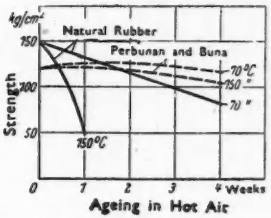


Fig. 1

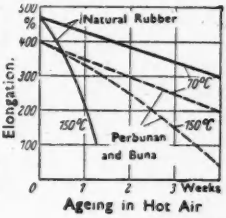


Fig. 2

For the application of soft grades of rubber certain principal precautions should be observed.

Under tension rubber tends to rupture, especially if exposed to light, oxygen, or ozone. Therefore, it is advisable to shape the rubber part accurately in order that it can be fitted without deformation and consequent stressing. Such precautions should also be taken when stressing is unavoidable, as, for instance, when laying cables where small radii of curvature should be avoided. Notches in rubber parts are distinctly undesirable, rupturing and tearing usually starts at these points.

One of the main applications for rubber is protective coating of parts exposed to corrosion or rusting. The rubber coating is made either by a spraying process or by laminar sheet covering. The former method is used up to 2 mm. thickness, while the latter from 2 to 8 mm. thickness. To increase the working life of the coatings, sharp extrusions and edges in the article to be coated should be avoided if possible. Welded seams should be ground or polished, and air pockets in the metal eliminated.

Another important field of application of rubber is for vibration damping and sound in-

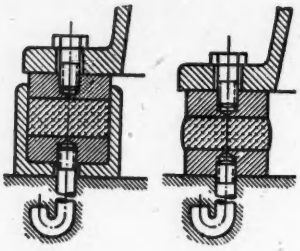
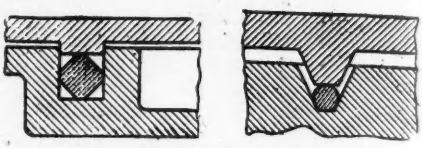


Fig. 3 (left). Wrong.

Fig. 4 (right). Correct.



Figs. 5 and 6. Correct design.

sulation (silencer, muffler). In such applications the rubber should only be stressed by a compressive force; tension or shearing forces cause tearing. A frequent cause of failure is that designers forget that rubber can change its shape but not its volume; this point is illustrated in Figs. 3-6.

For plates, rods, tubes, and protective covers hard rubber is usually employed. Hard rubber and Buna will withstand a temperature of 100° C., and show within this limit a remarkable resistance to attack by oils, acids, alkalis, steam, and gases. Petrol, benzol, and chiefly sulphuric carbonates on the other hand, attack rubber if given sufficient time.

The working life of rubber products depends mainly upon maintenance and the precautions taken when assembling. Rubber parts, as mentioned before, should be protected from light, heat, lubricants, liquid fuels, etc., if possible. Constant eccentric loading and buckling are also undesirable applications. If properly cared for, rubber parts retain their mechanical properties over a large number of years without failure.

SAFE SPEEDS FOR FLYWHEELS

By JOSEPH MARIN. (From *Machine Design*, Vol. 14, No. 1, January, 1942, pp. 41-44).

ROTATING discs and cylinders are examples of machine parts in which a combined state of static stress occurs. This article deals with the design of these elements considering the distortion energy theory and in some cases the usual shear theory. In addition, design charts are given which afford a convenient means of easily selecting the allowable speed for a disc.

Rotating Solid Circular Disc: a circular disc may be distinguished from a cylinder by the fact that the thickness of the disc is small compared to its radius, whereas in a cylinder the length is several times the radius. For the solid circular disc shown in Fig. 1, at any distance from the centre there are two components of stress as shown in the figure. The values of these stresses are

$$S_r = k_1 (a^2 - r^2), \quad S_t = k_1 (a^2 - k_2 r^2) \qquad .. \quad (a)$$

where $k_1 = 1/8 \ (3+m) \ (\delta\omega^2)/g$, $k_2 = (1+3m)/(3+m)$, in which m = Poisson's ratio, δ/g = mass per unit volume and ω = the angular velocity in radians per second.

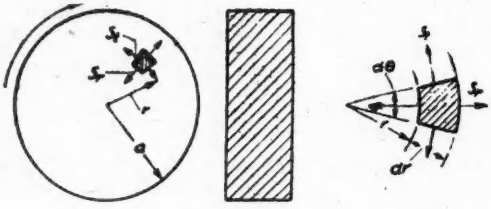


Fig. 1. Components of stress for an element of a solid circular disc.

Components become Principal Stresses.

Since there are no shearing stresses on the faces of the element shown in Fig. 1b, the com-

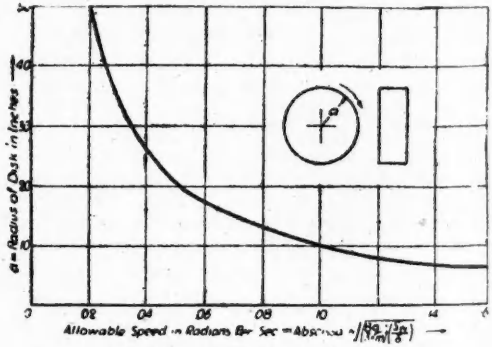


Fig. 2. Allowable speed for a rotating solid disc.

ponents of stress are the principal stresses. Then the maximum shear stress within the element is

$$S_s = \left(\frac{S_t - S_r}{2}\right) = \frac{k_1 r^2}{2}(1-k_2) = k_1 r^2 \left(\frac{1-m}{3+m}\right)$$

or

$$S_s = \frac{S_r}{2} = \frac{k_1}{2}(a^2 - r^2) \quad .. \quad .. \quad .. \quad (b)$$

or

$$S_s = \frac{S_t}{2} = \frac{k_1}{2}(a^2 - k_2 r^2)$$

Considering all possible values of r and m, the maximum value of the shear stress is, from these equations,

$$(S_s)\,\text{max} = \frac{k \,.\, a^2}{2} \quad .. \quad .. \quad .. \quad (c)$$

By the maximum shear theory, the allowable speed is obtained by equating the value of the maximum shear stress in equation (c) to the allowable value $S_w/2$ where S_w is the working stress in simple tension. This gives an allowable value for the magnitude of the speed.

$$\omega = \sqrt{\left(\frac{8g}{3+m}\right)\left(\frac{S_w}{\delta}\right)} \cdot \left(\frac{1}{a}\right) \quad .. \quad (1)$$

Fig. 2 shows the variation in the allowable speed ω for different values of the disc diameter in inches.

According to the distortion energy theory it will be necessary to consider the magnitude of the distortion energy for all possible values of r and to select the element for which this energy is a maximum. The expression for the distortion energy is

$$V = c\,(S_r^2 - S_r S_t + S_t^2) \quad .. \quad .. \quad (d)$$

where $c = (1+m)/3E$.

Substituting the values of stress components from equation (a) in equation (d), the distortion energy for an element at a distance r is

$$V = ck_1^2\,[(a^2 - r^2)^2 - (a^2 - r^2)(a^2 - k_2 r^2) + (a^2 - k_2 r^2)^2]$$

or

$$V = ck_1^2\,[a^4 - a^2(1+k_2)\,r^2 + (k_2^2 - k_2 + 1)\,r^4] \quad (e)$$

Maximum or minimum value of the distortion energy occurs for a value of r defined by $dV/dr = O$. From equation (e) this is for one of the following values of r,

$$r = O, \text{ or } r = \sqrt{\frac{1+k_2}{2(k_2^2 - k_2 + 1)}} \quad .. \quad (f)$$

Placing these values of r in equation (e), the maximum value of V is for r=O, and its magnitude reduces to

$$V = ck_1^2 a^4 \quad .. \quad .. \quad (g)$$

For simple tension the allowable value of V is

$$V = cS_w^2 \quad .. \quad .. \quad (h)$$

Equating the values of the energy in equation (g) and (h) gives the allowable speed as

$$\omega = \sqrt{\frac{8gS_w}{(3+m)\,\delta}} \cdot \left(\frac{1}{a}\right) \quad .. \quad (2)$$

This equation is identical with equation 1, as can be seen by noting that the two above theories coincide when the principal stresses at the critical element are tensile and equal in magnitude. For this particular case there is no difference between the values as given by the older shear theory and the newer distortion energy theory.

Rotating Hollow Circular Disc: For a hollow rotating disc, Fig. 3, the stresses at a distance r from the centre of rotation are

$$S_r = k_1\left(b^2 + a^2 - r^2 - \frac{a^2 b^2}{r^2}\right)$$

$$S_t = k_1\left(b^2 + a^2 - k_2 r^2 + \frac{a^2 b^2}{r^2}\right) \quad .. \quad (i)$$

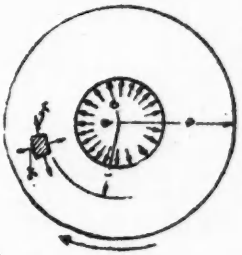


Fig. 3.

Stress on an element of a hollow disc.

where the values of k_1 and k_2 are as in the case of the rotating solid circular disc.

Determining Allowable Speed.

To determine an expression for the allowable speed in this case, only the distortion energy theory will be considered. The expression for the distortion energy in an element at a distance r is obtained by placing the values of the stresses from equation (i) in equation (d). Doing this, the expression for V reduces to

$$V = k_1^2 cb^4 \left\{[1 + (5 - 3k_2)\,\alpha^2 + \alpha^4] - (1+\alpha^2)\right.$$

$$\left.(1-k_2)\,x^2 + (1 - k_2 + k_2^2)(x^4) + \frac{3\alpha^4}{x^4}\right\} \quad (j)$$

where $x = r/d$ and $\alpha = a/b$.

In order to determine the critical element or the point at which failure would occur according to the distortion energy theory, it is necessary to

determine the maximum value of V. This can be done by using the condition for a maximum V; namely, $dV/dx=O$. Then the value of x for maximum V is defined by the equation

$$x^8 - 1.05\,(1-\alpha^2)\,x^6 - 4\alpha^4 = O \quad .. \quad (k)$$

This equation does not give the value of x for maximum V as normally obtained so that it is necessary to consider the variation in V as x varies. A graph showing this variation is given in Fig. 4. for a value of $\alpha = a/b = 0.5$. This shows that the

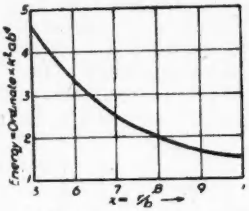


Fig. 4. Energy variation with respect to ratio r : b in hollow disc having inside to outside radius ratio of 0.5.

critical element is at the inner edge of the disc, that is, for a value r=a. At this point there is only a tangential stress so that the allowable speed can be obtained by placing the tangential stress S_t at r=a equal to the allowable stress in tension. Then from equation (i) the allowable speed is

$$\omega = \sqrt{\left(\frac{8_g\, S_w}{(3+m)\,\delta}\right)\left(\frac{1}{2b^2+a^2-k_2\,a^2}\right)}$$

or

$$\omega = \sqrt{\frac{8g\ S_w}{(3+m)\,\delta}} \cdot \frac{S_w}{b^2}\left(\sqrt{\frac{1}{2+\alpha^2-k_2\,\alpha^2}}\right)$$

Considering steel disc with Poisson's ratio m= 0.3, the allowable speed determined from equation 2 is plotted in Fig. 5 for various values of the ratio a/b.

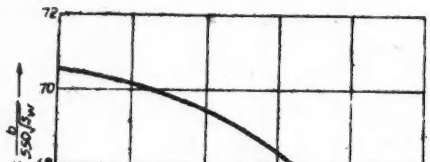


Fig. 5. Allowable speeds for rotating hollow disc.

Rotating Hollow Circular Disc with Boundary Stresses: Sometimes there are radial pressures

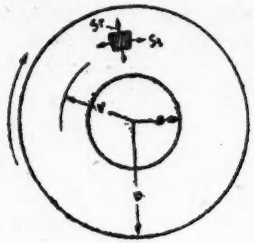


Fig. 6. Hollow disc with radial pressure caused by shrink fit on shaft.

or tensile forces distributed uniformly along either the inner or outer perimeter of the disc, Fig. 6. Examples of such forces are radial pressure from a shaft upon which the disc is shrunk, or a centrifugal pull from parts attached to the rim. Under these conditions the resultant stresses can be obtained by superimposing the stresses due to the boundary forces on those stresses due to the inertia forces. This procedure leads to the following expressions for the stresses

$$S_r = A + \frac{B}{r^2} - \beta_1\,\omega^2\,r^2 \quad .. \quad .. \quad (l)$$

$$S_t = A - \frac{B}{r^2} - \beta\,\omega^2\,r^2 \quad .. \quad .. \quad (m)$$

where $\beta_1 = (3+m)\ \delta/8g$, $\beta = (1+3m)\ \delta/8g$. In equations (l) and (m), the constants A and B are to be determined from the boundary conditions. Considering the case of a hollow circular disc shrunk on a shaft, the boundary conditions that the radial stress at r=a is -p and at r=b is zero give the necessary conditions for determining A and B from equation (l). Using these conditions, the values A and B become

$$A = \frac{1}{(b^2-a^2)}\,[\beta_1\,\omega^2\,(b^4-a^4) - a^2\,p] \quad .. \quad (n)$$

$$B = (a^2\,b^2)\left(\frac{P}{a^2-b^2} - \beta\,\omega^2\right) .. \quad .. \quad (o)$$

Stress value for a point in disc is now known in terms of the angular velocity and shrink-fit pressure. In order to determine the allowable speed it is necessary, as in the above problems, to find the position of the critical element. Using the distortion energy theory this element is the element for which the distortion energy is a maximum. The expression for the distortion energy for any element is obtained by placing the values of stresses from equation (l) and (m) in equation (d). These substitutions give a value of the distortion energy which can be simplified to

$$V = c\ \beta_1^2\,v^4\left[\left(\frac{1-k\,\alpha^2-\alpha^4}{1-\alpha^2}\right)^2 + \frac{(n-1)(k-\alpha^2+1)(3\,\alpha^2)}{\alpha^2-1} - x^2\,(n+1)\left(\frac{1-\alpha^2\,k-\alpha^4}{1-\alpha^2}\right) + \frac{3\alpha^4}{x^4}\left(\frac{k^2-\alpha^4+2\,\alpha^2-1}{\alpha^4-2\alpha^2+1}\right) + x^4\,(1-n+n^2)\right] \quad .. \quad (p)$$

where $v=\omega b$, $k=p/\beta_1 v^2$, $n=\beta/\beta_1=(1+3m)(3+m)$, $x=r/b$, $\alpha=a/b$.

In order to determine the maximum value of V in this equation it is necessary to consider all possible values of x. Using calculus for defining the maximum value of V, an equation of the eighth degree is obtained. Under these circumstances it is more convenient to determine the variation in the value of V for all possible values of x and then to select the maximum value of V. An examination of equation (p) shows that the value of x for maximum V will depend upon the magnitudes of p, v and α. For this reason it is desirable to substitute the numerical values of the constants in equation (p) to determine the value of x for maximum V as illustrated by the following example.

Considering a hollow steel disc with $m=0.3$, $\delta=0.284$ pounds per cubic inch, $p=5000$ pounds per square inch, $b=10$ inches, $a=5$ inches and $\omega=1800$ revolutions per minute; then $\alpha=0.5$, $k=p/\beta_1 v^2=8pg/\delta(3+m)v^2=4.67$ and $n=\beta/\beta_1=0.58$. Placing the values of these constants in equation (p), the distortion energy for an element at $r=bx$ is

$$V=11{,}250\,c\left(2{\cdot}37+0{\cdot}49\,x^2+0{\cdot}76\,x^4+\frac{7{\cdot}1}{x^4}\right)\ldots(q)$$

The variation of V for different values of x is shown in Fig. 7. For this particular example the maximum value of V is for $x=0.5$; that is, at the inner edge. To see whether this disc is safely designed it is only necessary to equate the energy in equation (q) for $x=0.5$ to the energy value in simple tension, which $V=cS^2$. This shows that the equivalent simple tensile stress $S=1150$ pounds per square inch. That is, the speed can be raised considerably over the value considered.

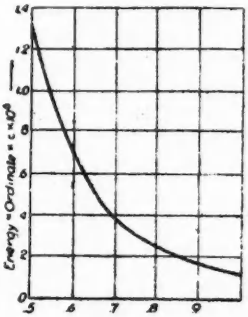


Fig. 7. Variation in distortion energy for disc with boundary stresses.

Critical Element is at Inner Edge.

If it is desired to obtain the maximum allowable speed for the above disc this can be done, assuming that the critical element is at the inner edge. Then equation (p) can be equated to the allowable value of $V=cS_w^2$ and with $x=0.5$ the value of k, and thus the speed ω, is determined. With this value of ω it is possible by again using equation (p) to find if the energy is maximum when $x=0.5$.

Rotating Hollow Circular Cylinder: If the thickness of a hollow disc becomes large compared to its diameter we have a three dimensional stress problem in which the three principal stresses, as shown in Fig. 8 are approximately

$$S_r=c_1\left(b^2+a^2-\frac{a^2b^2}{r^2}-r^2\right)\quad\ldots\quad(r)$$

$$S_t=c_1\left(b^2+a^2+\frac{a^2b^2}{r^2}-c_3r^2\right)\quad\ldots\quad(s)$$

$$S_z=c_2(b^2+a^2-2r^2)\quad\ldots\quad\ldots\quad(t)$$

where

$$c_1=\frac{3-2m}{8(1-m)}\left(\frac{\delta\,\omega^2}{g}\right)$$

$$c_2=\frac{m}{4(1-m)}\left(\frac{\delta}{g}\omega^2\right),\quad c_3=\frac{1+2m}{3-2m}$$

The above values for the stresses apply very well except for the parts near the ends of the cylinder.

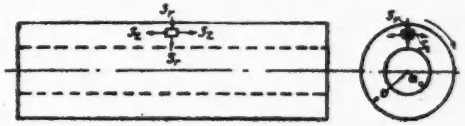


Fig. 8. Three-dimensional stresses on an element in a hollow cylinder.

In a manner similar to the above examples, the allowable speed can be obtained by first determining the element for which the distortion energy is a maximum. The value of the distortion energy for the three-dimensional case of stress is

$$V=c(S_r^2+S_t^2+S_z^2-S_rS_t-S_rS_z-S_tS_z)\ldots\quad(u)$$

Substituting the values of the stresses in equation (u) the distortion energy is

$$V=c\,b^4\left\{c_1^2[2+\alpha^2(7-3c_3)+2\alpha^4]-2c_1\,c_2(1+\alpha^2)^2-\right.$$
$$-x^2[c_1^2(1-\alpha^2)(1+c_3)+4(\alpha^2+1)-c_1\,c_2(5+5\,\alpha^2+$$
$$c_3+c_3\,\alpha^2)]+x^4[c_1^2(1-c_3+c_2^2)+2c_2(2c_2-c_1-c_1\,c_3)]$$
$$\left.+3\,c_1^2\,\frac{\alpha^4}{x^4}\right\}\quad\ldots\quad\ldots\quad(v)$$

where $\alpha=a/b$, $x=r/b$. To determine the value of x for maximum V it is necessary to know the ratio $a/b=\alpha$ and of m. With $\alpha=0.5$ and $m=0.3$.

$$V=c\left(\frac{\delta}{g}\omega^2b^2\right)^2\left[1.07-4.89\,x^2+0.041\,x^4+\frac{0.034}{x^4}\right]$$
$$\ldots\quad(w)$$

Variation of V with x shows that it is maximum for a value of x=1. The allowable speed is calculated by equating this maximum to the allowable value for simple tension equal to cS_w^2. Making this substitution the allowable speed is

$$\omega = 0.71 \sqrt{\frac{g S_w}{b^2 \delta}} \quad .. \quad .. \quad (4)$$

In a similar manner the allowable speed can be obtained for other values of x=a/b.

Allowable speeds for rotating disks of variable thickness can also be determined in a manner similar to the above. It is not convenient, however, in these more complicated cases to develop a general theory.

WAR-TIME HEAVY VEHICLE DESIGN IN GERMANY

By K. KUHNER. (From *ATZ, Automobiltechnische Zeitschrift*, Vol. 44, No. 23, December, 1941, pp. 592-597).

THE immense score of transport vehicles engaged partly in commercial traffic and partly in transporting troops and their supply to the various fronts has resulted in an urgent demand for lorries on an unprecedented scale. It is only natural, therefore, that development in design is closely related to new and changed requirements of war-time service.

During the last two and a half years, the development of heavy vehicle design in Germany took place according to the Schell-plan; its aim is first to increase the production by standardization, and secondly to permit conversion of the lorries manufactured, with a minimum expenditure of labour and material, to all-wheel drive, if this becomes necessary. The first aim could be easily attained by adopting only 6 standard types of lorries, namely 6½, 4½, 3, 1½, 1 and 0.65 tons capacity respectively. By this a large armada of lorry types has been eliminated which were formerly manufactured, not so much on account of the necessity for such, but because individual manufacturers wished to supply their own complete range of load carrying vehicles. As an illustration of the number of types eliminated by this standardization no less than 5 types were eliminated between the 3 and the 1½ types.

It was now a much easier task to complete the design of the few remaining types to suit mass production as is the case with the Opel 3 t., the N.A.G. Büssing 4½ t., and the M.A.N. 4½ t. types. The permissible overload that can be carried by these when converted to all-wheel drive is about 10% for the lighter types and 5% for the heavier designs. In general all types of lorries are more massively constructed than was the case before the war, this, of course, being due to the heavy strain to which they are put on the long supply routes, sometimes over tractless country.

The main problems of design were related to the switch over to substitute fuel and the reduction of speed in general to save valuable rubber by prolonging the working life of the tyres. Whilst so far as the change over from Diesel engines to solid fuel engines was only possible by a change over to the Otto cycle first, an elaborate and costly method, recently the problem has been simplified considerably by the pilot injection method, where small quantities of Diesel fuel are added to the main fuel. Not only does this solution offer technical advantages, but also the small reduction of power, unavoidable with Otto engines, is here eliminated.

The change over to producer gas fuel with Otto engines is less satisfactory, owing to the power reduction, even if the C.r. is increased. Practice has shown that the reduction of power is more marked with high speed engines having small cylinders than with slow and medium speed

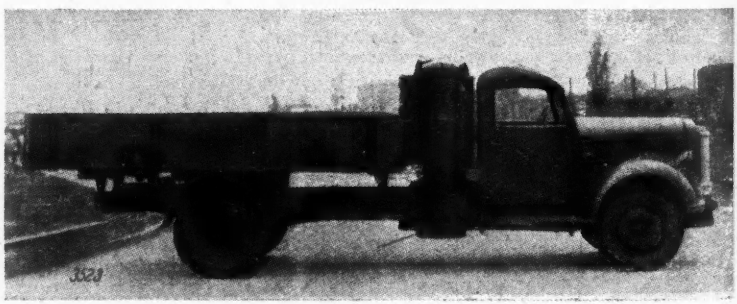


Fig. 1

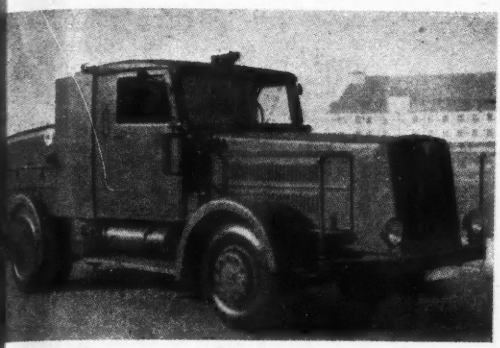


Fig. 2

engines with large cylinder dimensions. In order to compensate for this reduction of power and to maintain acceleration and pulling power, it was necessary to accommodate in the drive a some 15% larger reduction gear.

Modern research, however, attempted to overcome these difficulties by experimenting with various methods of maintaining power output, even under the changed conditions. So the firm Vomag has enlarged the bore of their engines from 130 to 140 mm. The power developed in the larger cylinders was now the same as when using petrol fuel. Again, other manufacturers attempted to find a solution by employing superchargers driven either directly by the engine or by an exhaust gas turbine.

No satisfactory solution has been found, so far, for the accommodation of the gas-generator, where it is accessible and yet not affecting the useful loading surface of the vehicle (Fig. 1). A more or less satisfactory solution is shown in Fig. 2, and for passenger cars in Figs. 3 and 4.

Fig. 3

The need for prolonging the working life of rubber tyres, as mentioned before, has induced engineers to specify maximum permissive speeds for the various types of heavy vehicles. As an example, the maximum speed for the 4½ t. vehicle was fixed at 40 km/h., and for the 6½ t. vehicles at 28 km/h. In other words, the maximum possible speed of the vehicles has been cut by approximately 40 to 50%. This speed control was a serious problem to the engineers; consideration had to be given not only to the simultaneous reduction in power by the change over to substitute fuels, but also to the problem of maintaining the efficiency of the car in post-war times. With Otto engines a reduction of speed was already caused by the installation of a larger reduction gear owing to the reduction in power. But when this speed reduction was not sufficient an additional speed governor had to be used, provided the smaller power output of the engine permitted this.

However, the bulk of transport vehicles now in service, in particular the Diesel engines, have a torque characteristic independent of the speed, and here a speed governor would inherently bring about a considerable reduction in power. To overcome this difficulty two alternative methods exist: the locking of the highest speed gear, or a speed indicator which operates when the maximum permissible speed is reached. From these the first method is most unsatisfactory as operation at lower than top gear decidedly increases the fuel consumption. Besides, in some classes, such as the 6½ t. class, the second highest gear would not result in a sufficient speed

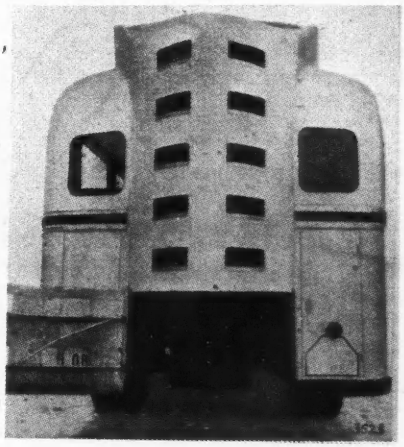


Fig. 4

reduction. More satisfactory proved to be the automatic speed recorder and indicator.

No final agreement has been reached so far as to the best type of engine for the individual vehicle classes. Nevertheless, Diesel engines are employed only for 3 t. and larger units, though their weight is about 50% more than of a corresponding spark ignition engine. For the 4½ t. class the engine is usually a six-cylinder, 105-125 h.p., engine of 7-8 litre stroke volume. Less uniformity exists in the 3 t. class. It appears that amongst the Diesel engines preference is given to the 4 cylinder type, owing to its shorter engine block and also because this simplifies production as the cylinder dimensions are identical with those of the six-cylinder type in the 4½ t. class. A great number of the 3 t. class, however, are still being fitted with spark ignition engines, these being of the same type as manufactured before the war for passenger vehicles. In this way the task of a special design of engine for mass production could be eliminated. In the heaviest class, that is, for the 6½ t. lorries, a six-cylinder 150 h.p. Diesel is used, developed from the 4½ t. class engine.

Extensive research work is being carried on by Fried. Krupp A.G. with a newly developed 2 stroke, 8 cylinder V. engine. The weight of this engine is about 33% less than that of a 4 stroke engine of similar output.

Meanwhile the development of change gears is proceeding uninterrupted by the war. The five speed gearbox developed from the four speed type by adding a fifth gear, is already universally adopted in the heavier classes of vehicles and its use is gaining ground in the smaller classes too.

A noteworthy solution towards increasing the speed range has been designed by the Faun works on their 150 h.p. engines. A four speed gearbox with approximately equal speed stepping is provided with an auxiliary gearbox. Thus 8 speeds can be obtained with approximately equal spacing over the whole range, which is a far more desirable feature than the previously customary 5 speed drive, especially on mountainous terrain.

A novelty in the field of power transmission is the so-called "distributor gear," which has been designed for the newly manufactured lorries which are suitable for conversion to all-wheel drive. Their duty is not only the distribution of power between front and rear wheels, but also to disconnect the front drive during travel over flat country. This gear has usually two speeds, engagement of the lower one simultaneously making the connection to the front wheel when the driving force has to be increased.

The simplicity of converting a vehicle with two wheel drive to a quadruple drive is shown in Fig. 5.

The final form of front wheel drive has also not yet been reached. The single-step power transmission with the large axle box requires not only adequate protection owing to its low position, but also an eccentric location to avoid striking against the crank case of the engine just above it (Fig. 6). A more sound arrangement is

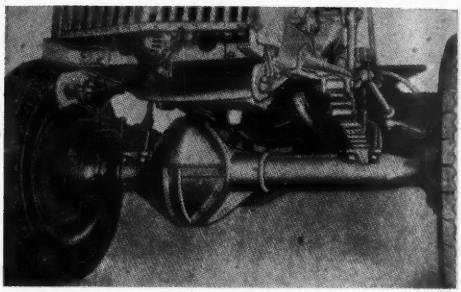


Fig. 6

to split up the power transmission into two stages, so that the diameter of the bevel gears is small enough to permit symmetric location.

The latest development tends to increase the versatility of heavy vehicles. The wheel base is being shortened to increase mobility on cross country routes, and with it the design had to be altered so that the engine extended into the driver's cabin, in order not to reduce the bulk carrying capacity.

The ideal solution, namely to shift the engine overall under the driver's seat, is only feasible with engines of low height. Such engines should have a very short stroke length with side valves, a feature characteristic to spark ignition. This solution is limited to light and medium tonnage lorries. Only when expensive engines of special design with horizontal cylinders are used, will the choice of construction be free; such development is, however, unlikely to take place within the next few years.

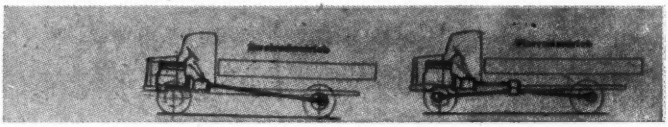


Fig. 5

UTILIZATION OF STEAM PASSENGER LOCOMOTIVES

By A. A. RAYMOND. (From *Mechanical Engineering*, Vol. 63, No. 12, December 1941, pp. 870-874).

(*Continued from page 77*).

While acceleration only has been shown, getting over the road in the minimum amount of time is affected also by the rate of deceleration, although in general more time is lost accelerating than decelerating. Fig. 4 shows the time that can be saved by proper braking and accelerating after a slow-down to thirty miles an hour.

It may seem that accelerating at maximum capacity is a minor item, but with long runs the number of accelerations necessary because of stops and slow-downs may, upon analysis, be found of such frequency as to make this an important item. On the New York Central, daily checks are made of tapes or charts from locomotives handling some 700 train divisions. The particular value of these charts is that instead of saying to the engine-man, "could you have accelerated at an increased rate ?" he can be told, "If your speed is 70 m.p.h five miles out, you can save two minutes on the running time." It is estimated that it costs a ton of coal to make up a minute by accelerating faster.

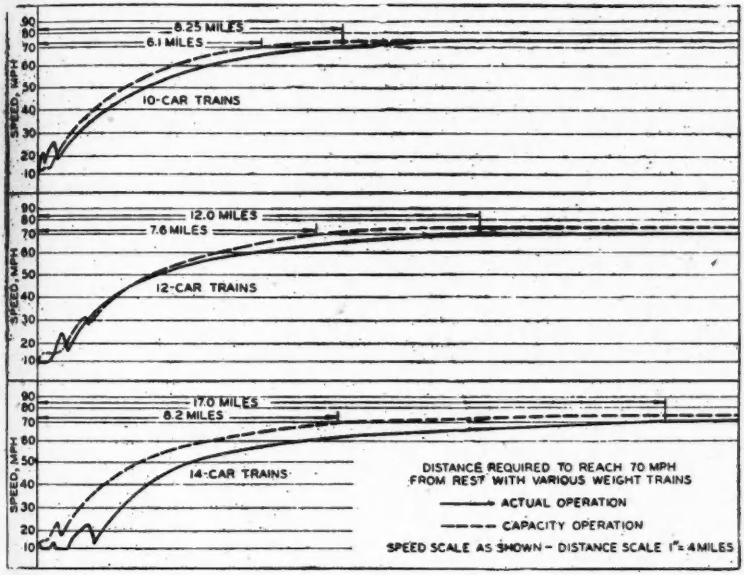
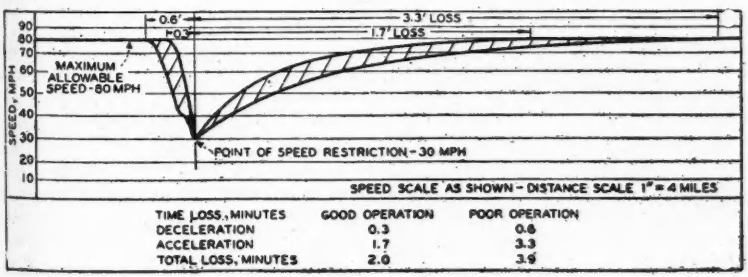


Fig. 3



TIME LOSS, MINUTES	GOOD OPERATION	POOR OPERATION
DECELERATION	0.3	0.6
ACCELERATION	1.7	3.3
TOTAL LOSS, MINUTES	2.0	3.9

Fig. 4

As the efficiency of the locomotive is improved, it seems almost necessary to study the operation of the different locomotives with a recording tape, thus determining the economical cut-off to use under different running conditions. After a train has reached full running speed the engine man should be encouraged to shorten the cut-off just as much as is practicable in maintaining the schedule. Education in the economical use of the improvements which are built into the locomotive is very essential.

4. Supplying coal and water for long high-speed runs. In general it is easier to supply water than coal, because water can, where available, be piped conveniently and arrangements can frequently be made to take water at regular stops where crews are changed, by having a supply and a standpipe large enough to fill a tender in the three minutes in which the train

should be standing still. Sometimes the lack of the right kind of water supply makes this difficult and expensive.

A study made for the last eleven years, Table II, of the average passenger-locomotive miles made per ton of coal indicates the increasing amount of coal required. These figures include both the coal used in road operation and the coal used at the terminals. This increase is the result of heavier

Table II.

Year	Average locomotive miles per ton of coal
1930	19.26
1931	18.85
1932	17.95
1933	18.30
1934	17.73
1935	17.43
1936	16.26
1937	16.12
1938	16.01
1939	15.55
1940	15.14

trains, faster schedules, air-conditioned equipment, and the like, and indicates what should be considered in determining ways and means of obtaining longer runs between coal stops or cut-outs for fuel, for higher utilization. In this connection attention is called to Fig. 5 which shows the coal and water capacities of the locomotive tenders under consideration. To the graph has been added tender proportions of the latest facility, a combination passenger and freight locomotive just recently put in service, the tender of which holds 43 tons of coal. The solid line in Fig. 5 connects the coal and water capacities of tenders actually in operation, while the broken line shows what the tender capacities would be if the ratio between coal and water capacities of the original class K-2 locomotives had been continued in the later class. It will be observed that the tendency is materially to increase the proportionate amount of coal carried. This trend in tender design has resulted from the installation of sufficient track pans with which to maintain service with a tender capacity of about 16,000 gallons. In other words, 16,000 gallons is enough for train operation between track pans.

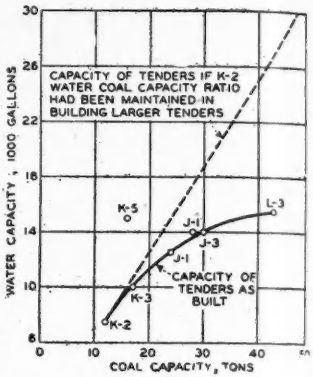


Fig. 5

As will be noted from the foregoing tables and charts, because of faster trains, increased train weight, and air-conditioning equipment, the fuel consumption for passenger locomotives is nearly a ton every 15 miles. Of course, we must realize that this figure covers the average for the year, whereas, the consumption is higher in winter and lower in summer. It covers all weights of trains, while more emphasis should probably be placed on the heavier trains which tend to increase the fuel consumption per locomotive-mile. It should be remembered also that the figure includes terminal as well as road fuel. But as a whole, the operation is getting close 15 road-miles per ton of coal for the principal trains, and it will be appreciated that in arriving at the final terminal there must be some coal left on the tender for emergencies.

THE STEREOAUTOGRAPH A6 OF THE WILD COMPANY

By Dipl. Ing. R. Voegeli.

(From *Schweizerische Bauzeitung*, Vol. 118, No. 10, September, 1941, pp. 120-122).

The A6 is a stereoscopic rectifier for making maps from aerial photographs taken approximately vertically as it is common practice in aerial photogrammetry.

Assume a section of a landscape being photographed from 3000 to 4000 metres twice within a short interval. The two point objects P and Q (Fig. 1) appear then as point images P', Q' and P", Q" respectively. O' and O" are the lenses in the respective instant of taking photographs, their distance b being called the base. If h is the flying height, the ratio b/h must be so chosen that the two pictures overlap one another by 50%.

The photographed landscape can be geometrically reconstructed whereby the base and flying height are adjusted to suit the scale of the plotting apparatus. Thus, we find the triangle O—O"P again in Fig. 2 and Fig. 3. The light rays P'O'P and P"O"P of Fig. 1 are represented by plotting rods L'O'P and L"O"P (3 in Fig. 2) which pass through the universal joints O— and

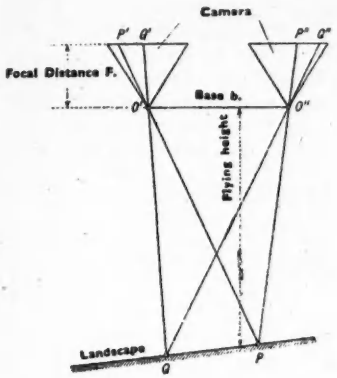


Fig. 1

O'. These enable the rods to be moved axially, too. P is fixed on the glass table (15, Fig. 2) and the rods form various angles when moved on the table by handle 14. Provision is made to vary the base b by altering the distance O'—O" between the limits of 70 to 250 mms.

To make room for the reflectors (6, Fig. 2) the images P' and P" do not coincide with L' and L", but are transmitted through these points by the "Nürnberger-scissors" (4, Fig. 2) as shown in Fig. 3. The middle point of the scissors

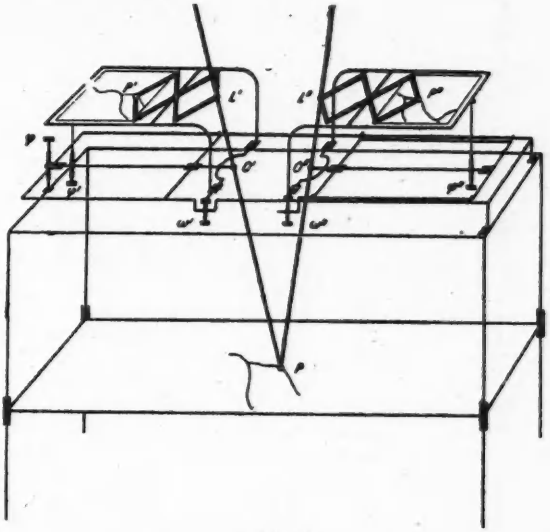


Fig. 3

is fixed, and L' and L" take the shape of sleeves 2 in Fig. 2.

P' and P" together with the co-ordinated point marks are projected into the telescopes 13. The vertical distance between O'L' or O"L" equals the focal length of the photocamera, and can be varied between 97 and 270 mms., by sliding of the photo-carrier, 7, along column 8. Owing to the unsteadiness of flight it is impossible to keep the axis of the camera constantly vertical. This fact is taken into account in the A6 instrument, insofar as the photo-carriers can be turned, each about three mutually perpendicular axes. The primary axes coincide with the direction of the base O'—O", and are called the lateral tilt, or ω axes (1 in Fig. 2). They are intersected by the secondary axes in O' and O" respectively. These are the fore-and-aft tilt axes or φ axes. They are hori-

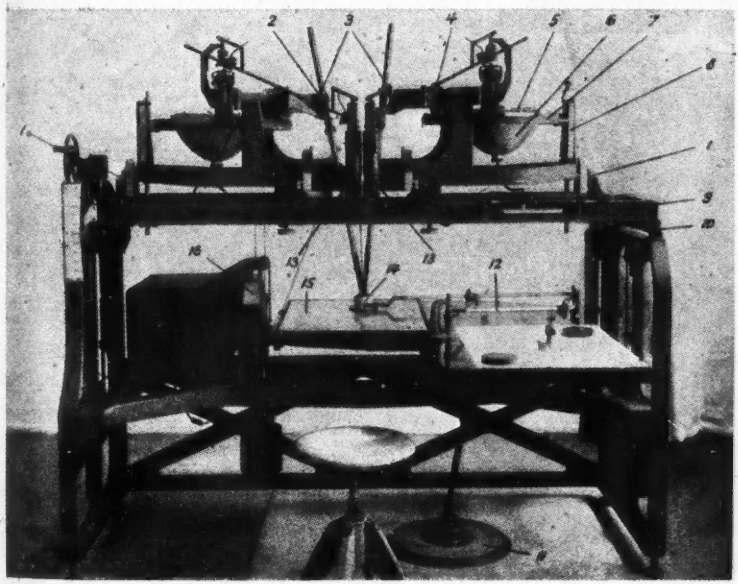


Fig. 2

zontal, but allowed to turn about the ω axes together with the photo-carriers. The third pair of axes, the x-axes, are perpendicular to the plane of the photographs thus representing the optical axes and enable the turning of the photos about their optical centre.

As it is impossible to maintain exactly horizontal flight, the base points O' and O" will always show slight height differences against one another. These differences are taken care of in the A6 by tilting the frame 9 about axis 10 together with the base and photos. All angles can be read off from suitable scales.

As already mentioned, the plotting rods at their intersection P rest on a horizontal glass table. This table is rigidly connected with the drawing board, and can be vertically lowered or raised by turning pedal 11. Thus a three-dimensional movement of the plotting rod handle 14 is made possible. The plane view of this movement is transmitted to the drawing paper by means of the pantograph 12. The ratio of the transmission from handle to tracing pencil can be varied continuously between 1:1 and 20:3. The vertical adjustment of the drawing board, or what it amounts to, of point P can be read off from a scale which is projected to the glass-screen 16.

The task is now to give a couple of photographs in the A6 the same absolute position in space they had at the moment of exposure. This task can be solved in four steps, as follows: (1) The inner orientation of the photo-camera, which is the introduction of the focal length of the camera into the rectifier; (2) The mutual orientation of a pair of photos consisting in giving the photos the same relative position to one another in the rectifier as they had at the instant of exposure. As a criterion for this agreement the so-called y-parallax is used. (A full account of the theory of the y-parallax is given in " Lehrbuch der Stereo-Photogrammetrie," by Baeschlin & Zeller); (3) The variation of the base. To know the true base b, the distance of two point objects P and Q (Fig. 1) must be known. Two such known points must be checked on a sheet of paper by means of the pantograph, and if there is no agreement, the base adjusted accordingly; (4) The model-swing. Finally, the photos must be swung about two axes, so that their plane view in the rectifier has the same position as that from the air relative to the horizon.

This can be visualized on hand of Fig. 1. If one turns both cameras together with the base say about mid-base, the whole system of rays turns with them, thus the intersection points P and Q too. From the vertical distances of these fixed points the position error of the cameras and of the base can be found.

The A6 is not only cheap but for its price very accurate, and owing to its simplicity well suited for instructional purposes. As an example a survey at altitude of 3500 metres is plotted with A6, and shown in Fig. 4.

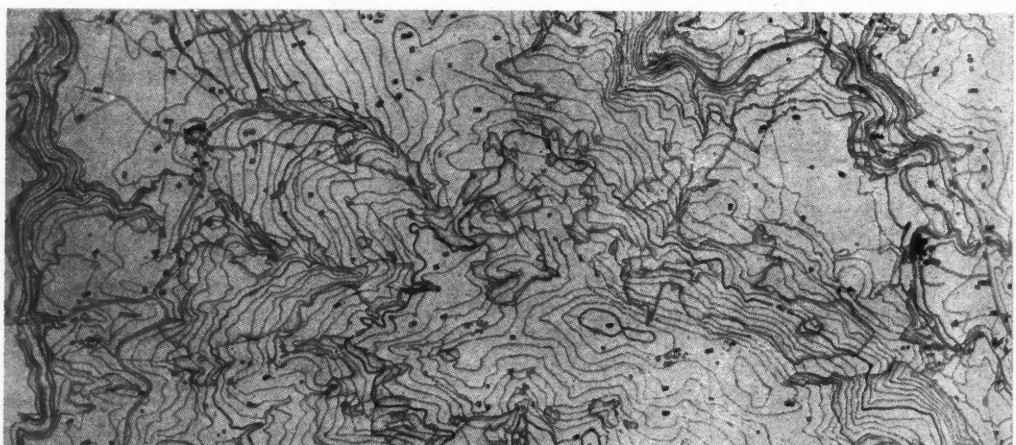


Fig. 4

MECHANICAL-OPTICAL EXTENSOMETER

By HEINRICH FREISE.

(From *Zeitschrift des VDI*, Vol. 85, No. 47/48, November, 1941, pp. 919-921).

THE stress distribution under static loads in construction parts is important information, and can be best obtained by measurements with an extensometer. The base length, i.e., the length of material over which extension measurement is made should be short, so that peak stresses also can be determined. A recently designed extensometer is described in the following, which, being of the revolving mirror type, gives very accurate results, and is not greatly affected by rough usage.

Figs. 1 and 2 show clearly the working principle of the instrument. At the bottom of the tube (a) a fixed knife edge (b) is formed, a second knife edge (c) being in rigid contact with the mirror (d) which is pin pointed to the tube (a), so that rotation of the mirror through a small angle is possible. The base length is 20 mm., as confined by the two knife edges. At the top of the tube (a) a glass plate (e) is fixed together with a reflecting prism (f). The prism projects the image of a scale through the plate on to the mirror, from where it is reflected at some angle, depending upon the angular position of the mirror, and is focussed back on to the scale which lies in the focal plane of the objective (g). The latter is closely positioned to the mirror. The reading (y) between the datum on the plate and its image projected by the mirror corresponds to a definite rotation of the mirror through φ degrees. Readings on the scale are made through an eye piece (h).

If the elongation measured across the base length is equal to x, then for small values of φ, $y/x = 2f_1 z$, where f_1 is the focal length of the objective (g), and z is the height of mirror above the surface of material under examination (Fig. 1). The total magnification is then given by

$$V = 2f_1 . (v/z)$$

where v is the magnification of the eye piece given by $v = S/f_2$, S being the focal length of a healthy eye and f_2 that of the eye piece. (S is actually 250 mms.).

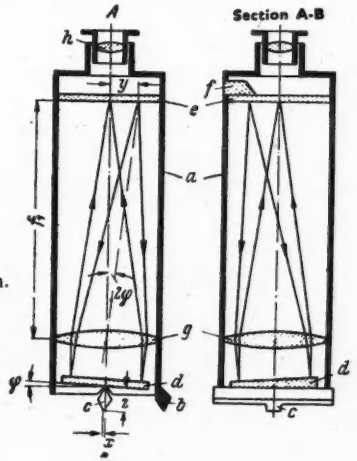


Fig. 1
(a) main tube.
(b) fixed knife.
(c) movable knife.
(d) mirror.
(e) glass plate.
(f) reflecting prism.
(g) objective.
(h) eye-piece.

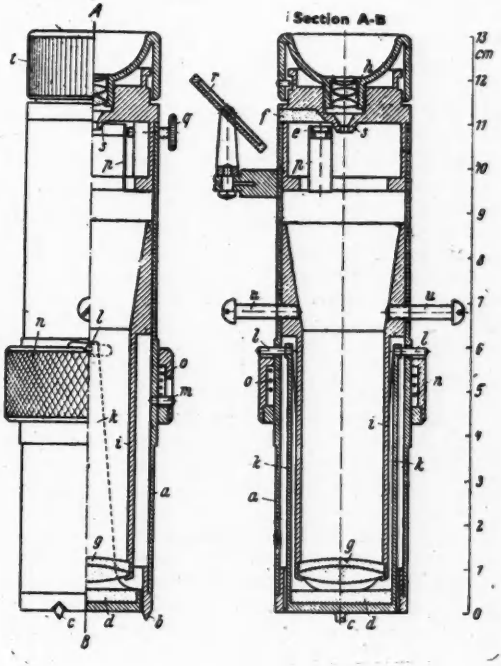


Fig. 2 (right): (a) to (h) as in Fig. 1; (i) tube holding objective; (k) arms; (l) trinnions; (m) guiding pin; (n) guiding ring; (o) spring; (p) leaf spring; (q) adjusting screw; (r) adjustable mirror; (t) microadjuster; (u) holding bolts.

It is necessary to bring the head close to the tube when taking readings, and in instances when this is not possible, the image of the scale and the datum is projected to a screen. The total magnification is easy to calculate.

Construction of Instrument.

This is shown in Fig. 2. The fixed knife edge (b) is a segment of a ring fitting into the main tube (a). The objective (g) is mounted in the main tube by means of a concentric tube (i). Two arms (k) pivoted at (l) are placed in the annular space between the tubes and carry at the lower end the reflecting mirror and the movable knife edge (c). A ring (n) enclosing the tube (a) is forced against the trunnions (l) by the spring (o) so that before placing the instrument in position the levers (k) and hence the mirror is brought to zero position. After attachment the ring is pushed down and secured by a bayonet catch, permitting free rotation of the trunnions (l).

The glass plate (e) is mounted between two leaf springs (p) and presses against a screw (q), which serves for zero adjustment. Close above the plate is a reflecting prism to illuminate the scale. An adjustable mirror (r) throws light on to the prism.

The total weight of the instrument is 127 kg. Attachment to the work is as shown in Fig. 3.

Accuracy.

The instrument was calibrated and checked on a Huggenberger calibrator. Over the whole range the magnification was found to be the same and independent of the pressure of attachment, which was between 0.5 and 2.6 kg. The total scale range corresponds to an elongation of $\pm$ 63.2μ. This range can actually be doubled if readings are started from one end instead of from the centre.

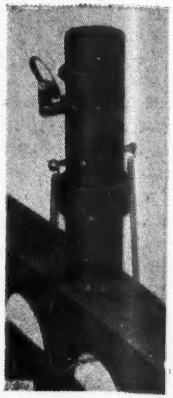


Fig. 3

The mean error of measurements was estimated to be about $\pm$0.13μ, which about 0.1% of the total scale reading. This accuracy is generally sufficient, and is the same for direct observation of projection on to a screen.

GAS-FILLED CABLES

By E. DORFEL. (From *Zeitschrift Deutscher Ingenieure* [*VDI*], Vol. 85, No. 20, May 17th, 1941, pp. 463-6).

FOR very high voltages, cables with a homogeneous dielectric have been used instead of cables with compounded paper insulation. Most of the cables for more than 60 kV are oil-filled cables. Gas pressure cables are another type of cable for extra high voltages. This type of cable is laid in steel conduits filled with gas at a pressure of 12 to 15 atm., which exerts its pressure to compress the cable as a whole. In both systems reliability and capacity, greater than that of compounded paper insulated cables, is based on the prevention of the formation of voids in the dielectric, in which ionisation may occur. However, the costs of the oil storage tanks and the other accessories of the oil cable plants for 100 to 150 kV amount to 15 to 20% of the cost of the cable. Such expenditure may be tolerated with cables for very high voltages. But, though superior to mass impregnated cables in regard to electrical reliability, they can not be used, for economical reasons for medium voltages oil cables. The situation is similar with gas pressure cables in consequence of the high cost of the steel pipe line.

Therefore the great interest for the design of a new type of cable, which could be rated higher than the compounded paper insulated cable, the rating of which is limited by the admissible temperature rise, but which would combine low costs with the reliability of the oil-filled cables attained by the exclusion of any voids in the dielectric. A solution of this problem has been found in cables, the dielectric of which is filled with gas under such a pressure, that glow discharges cannot occur under normal working conditions.

The proposition to fill paper insulated cables, impregnated or not impregnated, with gas under pressure was not new. It had been made by FISHER and ATKINSON in 1927 (they proposed a pressure of 100 lb. p. sq. in), BEAVER 1929 and DUNSHEATH. All these designers used very high pressures (15 atm.) as necessary with cables for very high voltages, which they endeavoured to develop. However, it seems not yet quite evident, if it is possible to make a gas-filled cable for extra high voltages, which is equivalent to an oil-filled cable.

Cables for Medium High Voltages.

The situation is very different in regard to voltages of 20 to 60 kV. With such cables, even when carrying high loads, reliability can be attained using moderate gas pressures (2 to 4 atm.). Based on extended research and long laboratory reliability tests a type of a gas-filled cable has been developed by the A E G about 2 years ago, and one cable of this type for 30 kV, 3×120 and 3×185 mm² cross section has been laid in Germany, part of it as river cable.

The cross section of a gas-filled three-core cable (Fig. 1) resembles that of a so-called H-cable. The triangular spaces between the cores which may be wormed or may be left empty, are used for feeding gas under pressure into all voids of the insulation, which developed during manufacture or may have been produced through repeated heating and cooling. The insulation of the cables consists of carefully lapped tapes of special paper, thickness of the lead sheath has to be made appropriate to the deformations of the cable which may occur during laying. The sheath has to give a good support to the double threaded protective armouring which is able to withstand a multiple of the gas pressure. The insulation of gas-filled cables can be made much thinner than that of mass-impregnated cables, which is liable to be deteriorated by ionisation in voids. In order to prevent movements of the compound, special materials are added into the stranded conductors, which give to the compound a higher viscosity. When impregnated the cable is filled with a neutral gas, e.g., nitrogen. Pressure is made so high, that no glow discharge can occur at normal voltage, for voltages of 20 to 60 kV, 2 to 4 atm.

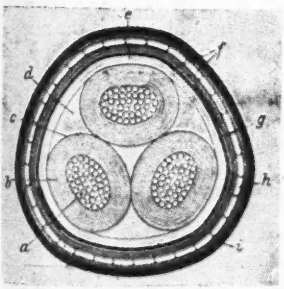


Fig. 1. Gas-filled cable, 3×120 mm², 30 kV: (a) conductor; (b) paper insulation; (c) metallised paper; (d) gas-filled interstices; (e) lead sheath; (f) paper with bitumen; (g) iron-tape armouring to withstand the pressure; (h) flat wire armouring to stand the stresses during laying; (i) jute serving.

Fig. 2 shows the relation between pressure and the beginning of glow discharges. The curves are calculated for voids of various thickness using PASCHEN's law, but applying some simplifications. Checks on cables showed that the discharges began at an even higher voltage than given by the the curves, probably the suppositions made for the calculation being too unfavourable. Pressure cables may carry 20 to 25% higher current than compounded paper insulated cables, and a permanent temperature rise of 40° C instead of 25° C can be permitted, as the insulation does not deteriorate by inner discharges even when subjected to heavy strains by frequent heating and cooling.

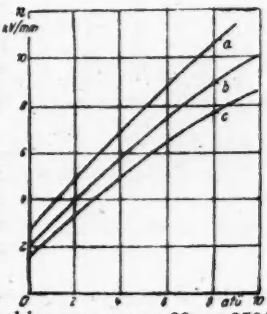


Fig. 2. Beginning of glow discharge in cables with pressure gas filling. Curves calculated for air gaps of 0.05 mm. (a) 0.1 mm. (b), and 0.2 mm. (c)

The impulse strength of the gas pressure cable is a little lower than that of a new compounded paper insulated cable for the same voltage, because a thinner insulation may be applied to the gas-filled cable, which undergoes no deterioration like the compounded cable.

Tests during manufacturing are made without gas pressure. Capacity and tan δ are measured with the Schering bridge. Then the cable is tested with d.c. voltage of six times the rated voltage between the phases. A shock test may be made before or after the d.c. voltage test. One

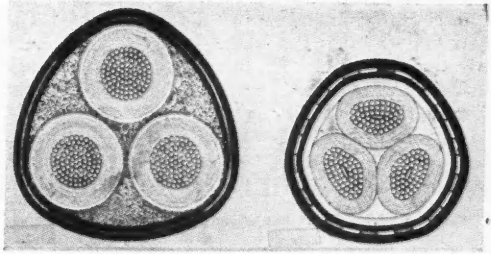


Fig. 3 Fig. 4

Fig. 3. Compounded paper cable 3×300 mm².
Capacity of transmission approximately 18,500 kVA.
Weight of lead approx. 9.6 t/km.
Weight of cable approx. 21.3 t/km.
Weight of conductors approx. 2.5 t/km.
Weight of steel armouring 4.3 approx. t/km.

Fig. 4. Gas-filled cable 3×185 mm². Capacity of transmission approximately 18,500 kVA.
Weight of lead approx. 7.6 t/km.
Weight of cable approx. 15 t/km.
Weight of conductors approx. 1.6 t/km.
Weight of steel armouring approx. 2.5 t/km.

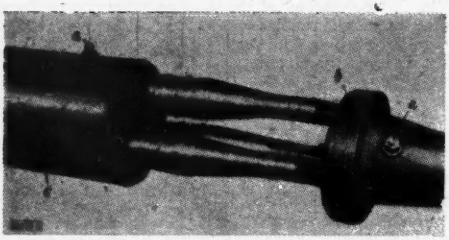


Fig. 5. Straight through box for gas-filled cable (a) valves.

sample is subjected to a bending test and insulation is tested again afterwards by d.c. voltage or by shock. After laying, the whole run inclusive of fittings is tested with gas of 1.5 times normal pressure and tan δ is measured under normal pressure in dependance on the voltage. A gas-filled cable is considered as sound, if up to 1.25 normal voltage between phases no glow discharge is started.

Considerable savings on metal can be achieved with gas-filled cables as can be seen from the comparison of a compounded and a gas-filled cable of the same transmission capacity (Fig. 3 and 4). These savings are much greater than the additional costs involved by the gas filling of the cable. It may be found astonishing, that for voltages up to 60 kV cables with shaped conductors are used. This can be done, as a special method is used for lapping the paper tapes, which avoids any crumbling of the paper.

The fittings for the new type of cables were developed from those of oil-filled cables. The straight through box (Fig. 5) consists of a very strong steel pipe. It contains a valve on each side by which the connexion of the inside of the box with either cable can be interrupted separately. A trifurcating box is shown in Fig. 6. Gas is fed from steel cylinders c (Fig. 6), equipped with safety and pressure reduction valves placed at both ends of long runs of cables, and at only one end of short ones. Pressure is controlled by a reducing regulator.

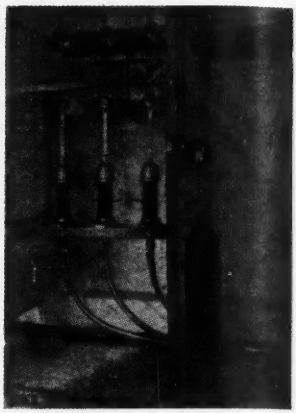


Fig. 6. Trifurcating box for gas-filled cable.

It is an important and very advantageous feature of the new type of cables, that they do not need any apparatus located along the run of the cables and that even long runs are fed only from the ends. No other subdivision of the runs is necessary except the valves in the normal fittings. Bulky installations like those necessary with oil-filled cables for the expansion of the oil and for producing the necessary pressure can be spared.

The measurements made after the installation of the cable showed a constant tan δ up to full voltage. The pressure inside the cable preserves it from the entrance of moisture should the lead sheath be damaged. Any leaks are detected at once by the fall of the pressure, so that the necessary remedy measures can be taken, before an earth fault or a short-circuit may occur. The failure can be sought on the live cable. For this purpose the valves in the fittings mentioned before are very useful.

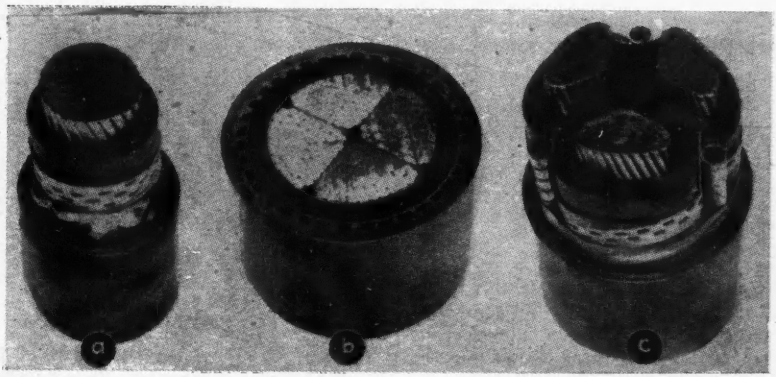
GAS-FILLED CABLES AIDS SYSTEM SUPERVISION

By E. R. THOMAS, Consolidated Edison Co., N.Y. (From *Electrical World*, Vol. 116, No. 10, September 6th, 1941, p. 47).

THE possibility of constant supervision over the the tightness of a low pressure gas-filled cable system has been considered as an outstanding advantage of this type of cables. Fig. 1 shows cross sections of various cables. In the single conductor, 15 kV gas-filled cable (a) manufactured by General Electric Co. the embossed spacer tape provides a gas channel between sheath and cable insulation. Fluted lead in (b) manufactured by General Cable provide the gas channel in this single conductor 15 kV cable. The three-conductor 500.000 circ. mil. 27 kV cable (c) manufactured by General Electric Co. resembles oil-filled types except that the channel spaces are filled with nitrogen under 10 lbs. pressure. Operating voltage stress in Volts per unit is considerably higher than in compounded cables. Gas-filled cables require therefore, less insulation for a given conductor size than solid types, hence permit additional capacity in existing ducts.

Pressure on the cable system normally shows a variation from 10 to 12 lbs. in changing from no load to full load. Fig. 2 is the view of a feeding point, right to left, are pressure alarm relay, high

Fig. 1



pressure bottle of nitrogen with reducing regulator and a graphic meter temporarily installed to provide a check on the cable operation.

The Consolidated Edison Co., N.Y., laid 28.500 feet 3-core 15 kV cable in June, 1938, adding 40.000 feet of 27 kV cable about 1 year thereafter. The longest operating feeder measures approximately 7500 feet. Up to now there have occurred no electrical failures in the cable system. There have been a few gas leaks indicated by the alarm system. Some of these have been due to gas leakage at fittings and two cases to damaged cable sheaths in a duct. Another leakage had been caused by a compressed air drill. But there resulted no electrical failure.

Over all costs of gas-filled cables are somewhat higher than for solid type cables. But it is believed, that the new type possesses operating advantages, which would tend to reduce electrical failures.

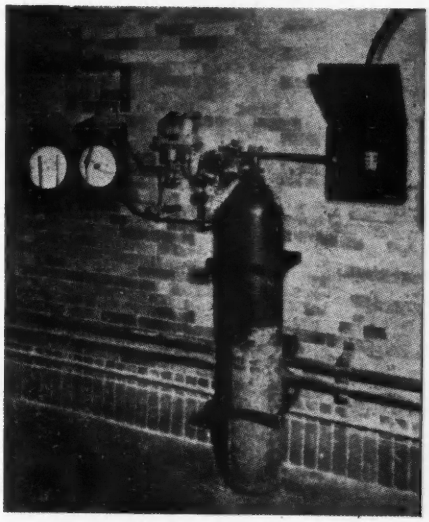


Fig. 2

HIGH SPEED PHOTOGRAPHIC ILLUMINATION

(From *Elektrotechnicky Obzor*, Vol. 30, No. 8, February 21st, 1941, according to J. VASTMANN, *Zeitchr. für Techn. Physik*, Vol. 21 (1940), p. 223).

THE requirements for the photography of rapidly moving objects consist of a camera and an illuminating source emitting intensive flashes of light of small duration, the camera shutter remaining open. The source is formed by a three-electrode spark-gap connected to a condenser C (Fig. 1), which is charged to such a voltage as will just not cause a disruptive discharge in the gap. A voltage impulse on electrode 3 breaks the electrostatic equilibrium in the gap and a spark flashes between the main electrodes. This spark is delayed for a certain time after the impulse has been impressed upon the primary of T, and this lag is not constant, but shows a certain dispersion. By an appropriate design of the gap, the average interval of the time between the initial impulse and the spark is kept under 0.25×10^{-6} secs.

In order to accumulate a large amount of energy, a high voltage would be the most favorable, but only 6 kV are used, in order to reduce the incumbrance and to have a spot-like source. Indeed, as the length of the spark is about 8 mm. and its diameter some 3 mm., the surface area of the source is only 0.25 cm^2.

The current in the spark is a damped A.C., as

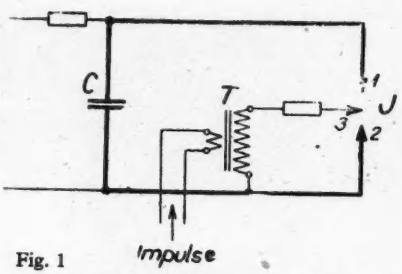


Fig. 1

the circuit contains inductances. If the transformation of the energy stored in the condenser into energy of light is to have a good efficiency, the ohmic resistance of the rest of the circuit must be small compared to that of the gap. This is obtained by means of short connections with large cross-sections.

Next to the intensity of the light, the duration of the spark is of importance. Every half-wave of current generates a corresponding flash of light, and certain electrodes (copper, iron) give a constant component, due to radiation from the metallic vapours. By this effect the flash is prolonged up to 10^{-3} secs. and more, and, in consequence, tungsten electrodes have been used. Concentration of the light into the first half-wave has been obtained by means of short, parallel, connections close together, and reduction of capacitance in the circuit. Fig. 2 shows an oscillograph of the intensity of light. Only the first half-wave has a photographic effect. The efficient spark lasts about 2.10^{-6} secs., this means for an object with a velocity of 100 m/sec. and a representation of 1:5 a sharpness of 0.04 mm.

The maximum intensity of the light, normal to the centre-line of the gap, is about 2.10^6 HJ, and the maximum brightness is about 8.10^6 Sb. Fig. 3 shows the arrangement of the whole apparatus. The extinction of a neon lamp in the condenser charging circuit after some 20 secs. indicates the end of the charging. The circuit of condenser C_1 is the same as shown in Fig. 1. The impulse for the discharge is given by the thyratron Th, which is started by a voltage impulse of at least 30 V. on its grid, introduced at V. To direct the light as required, an incandescent lamp L_2, the wire of which has the same surface area as the spark, is put in its place.

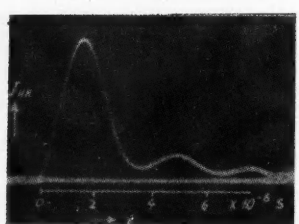


Fig. 2

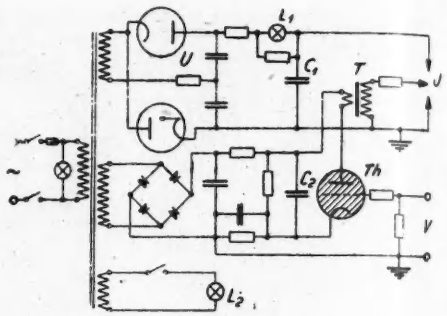


Fig. 3

Fig. 4 shows a calibre 5.6 mm. bullet, muzzle velocity 360 m/sec., after passage through a 1 mm. steel plate. The spark has been started by the bullet tearing off a tin-foil ribbon and cutting in this way an electric circuit. The apparatus may also be used for photographing periodic phenomena, where usually only some part of the movement is of interest. In such a case it is governed by a stroboscope, through which the object is observed. The electrical impulses are synchronous to the intervals when the object appears in the stroboscope, and are conducted to the apparatus through a press-button switch. After pressing the button the first electric impulse starts the spark. If a neon-lamp stroboscope is used, this device becomes particularly simple.

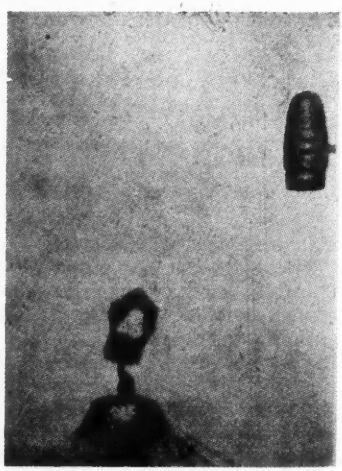
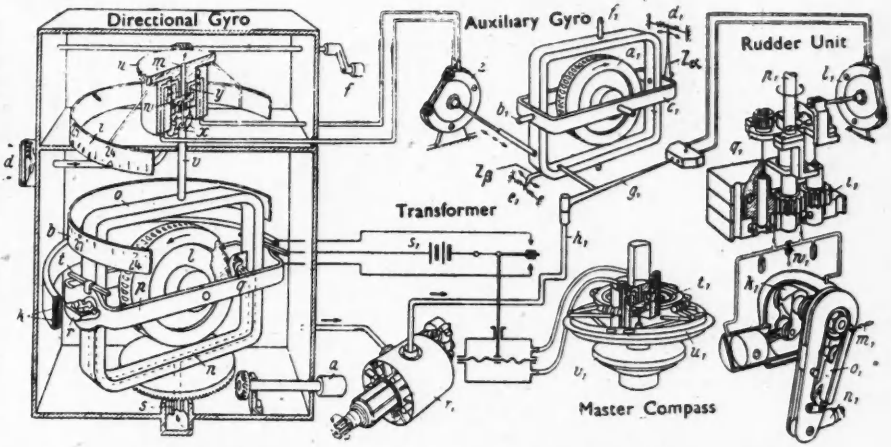


Fig. 4

AUTOMATIC DIRECTIONAL CONTROL

(From *Schweizerische Bauzeitung*, Vol. 118, No. 13, Sept., 1941, pp. 150-151).



THE aircraft magnet c compass gives reliable directional guidance, but the oscillations of its magnetic system about the north-south axis cannot be kept within desired limits.

On the other hand the axis of a gyroscope remains "fixed" in azimuth; being, however, subject to a slight precession.

According to an article by G. Wünsch in the *Zeitschrift of the VDI* a combination of the compass with the gyro in a self contained automatic control unit has been accomplished so that the deficiencies of the single elements are eliminated. Thus the magnet system is used to reset the gyro which gives impulses to the rudder. If the magnet system deviates from the longitudinal axis (direction of flight) of the aeroplane by the angle φ, and this deviation does not equal the desired course x, the gymbal ring, p, will turn about its axis, r, according to the sign of the difference $\varphi - x$. (The "command" $\varphi - \dot{x}$ of the compass is transmitted by electro-pneumatic means to the coils, k, which generate a magnetic field, acting on the permanent magnets, t). To a turn about the axis, r, the gyro responds with a turn about the axis, v, a motion which represents a second "command"; this being transmitted pneumatically to the rudder. As long as the angle Ψ between a perpendicular to the plane of gymbol, o, and the direction of flight does not differ from x, i.e., $\Psi = x$, the rudder will not be deflected. Oscillations of the magnet system have no influence on this combination of compass with gyro, as a mean difference between φ and x other than zero can overcome the inertia of the gyro only and cause thus rudder deflections.

Obviously the arrangement so far is inadequate because as long as the rudder is deflected so that the aircraft alters course by a right-hand turn, the aircraft not only remains in a right-hand turn till the true course is reached, but exceeds that desired value by an amount necessary to give impulses in the opposite direction. Thus, instead of keeping a straight line the aircraft would make turns about the yawing axis with angular speed, ω, and of alternating direction. To obviate this, the rudder-command is made dependent on the angular acceleration, $\dot{\omega}$, by using an auxiliary gyro, as shown in the figure. The axis, f_1, of the inner gymbal ring is parallel to the yawing axis. Under the action of spring, e_1, the gymbal ring is forced to turn about axis f_1, with the same angular speed, ω, as the aircraft when in a horizontal turn. In response, the gyro precesses, and gymbal, c_1, turns about the axis b_1, until the turning moment equals that induced by the pointer Z_α, under the action of spring, d_1. If the aircraft turns with a constant ω, the pointer, Z_α, indicates the rate of turn of the aircraft which is proportional to the pointer deflection, α:

$$\alpha = \frac{C}{k} \Omega \omega \quad .. \quad .. \quad (1)$$

Ω is the angular speed of the gyroscope, C its principal moment of inertia, k a constant. If ω is not constant but varying with time, $\frac{d\omega}{dt} = \dot{\omega}$ will be indicated by the pointer Z_B. The deflection B of Z_B is thus given by:

$$B = c\,\dot{\omega} \quad .. \quad .. \quad (2)$$

The values $\varphi - x$, ω, $\dot{\omega}$, are transmitted to the tube g_1, which is brought into one of the three positions "left," "right" or "middle." The "command" is then executed through l_1 to q_1, which switches the oil-pump to k_1, thus actuating the rudder by means of n_1. For the rest the figure is self-explanatory.

stripped water

In the preparation of tobacco for good quality cigarettes, the centre stem is stripped from the leaf in the manner shown above, but if this process is not carried out, tobacco including the centre stalk is uneven and contains dust which irritates the throat of the smoker. Similarly with the water required for feeding into water tube boilers it is necessary to strip from it the soluble salts and other impurities in order to secure a non-scaling, non-corrosive feed water. The Permutit "Deminrolit" Process produces a perfect stripped water from any supply without the use of heat or steam.

For full particulars write for publication "Distilled Water without Distillation" to The Permutit Company Limited, Dept. T.B., Permutit House, Gunnersbury Avenue, London, W.4. Telephone: Chiswick 6431.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- PERSONAL
- NOTES

NEW EQUIPMENT

NEW ROLE FOR STEAM PLANT.

Portable Instantaneous Generator Unit.

INSTANTANEOUS steam raising plant, besides filling its peacetime role of providing steam heat for Diesel rail cars and passenger aircraft may now have a new part to play, that of decontamination of war gases from machinery, road transport vehicles and even aircraft.

An interesting example, suitable for this class of work, is the Vapor-Clarkson Generator, which is an automatic oil-fired steam generator of the continuous coil water-tube type. Its great advantage for the work described above is the extreme compactness and portability of the unit. The stationary model occupies no more than three feet by two feet, excluding motor, and the portable model, complete with fuel and water tanks is mounted on a trailer no more bulky than the light trailer pump used for fire-fighting.

Not only can this Vapor-Clarkson Generator supply both dry or superheated steam, but it can produce high-temperature liquid sprays, at either high or low pressures. Thus it can be used for many diverse purposes in industry, and even cover several widely different jobs in the same factory.

The standard unit has a capacity automatically variable from 100 to 300 lbs. of steam per hour of dry or superheated steam at pressure up to 200 lbs./sq. in. : superheat is adjustable up to 500°/600° F. As the unit has a heat release of over a million B.T.U.'s per cubic foot of combustion space, an unusual amount of steam is delivered in relation to size.

The generators are constructed with a patented staggered coil assembly, which can be withdrawn from the unit without trouble by a single operation, either for cleaning or for replacement of a new coil unit. The whole of the conversion from water to steam is effected within the compass of this coil, so treated feed water is essential. The coil is fully compensated for expansion and is designed to split the combustion gases into narrow layers. As these gases flow in the opposite direction from the flow of water in the coils, high thermal efficiency results.

Only solid construction will produce such an unusual performance from so compact a unit, and so the coil is made from open hearth steel tube, the body and fire components from close grained cast iron and certain auxiliaries from bronze.

Component parts are chosen with care, having regard to the many exacting duties of the generator. Tecalemit filters are used in the oil feed and a Waynes "Viking" fuel pump controls the supply of the fuel. This pump, the Duplex feed water pump, the ignition magneto and the blower fan for combustion air are driven by one unit, which may be either an electric motor or internal combustion engine. The driving assembly is thus somewhat complicated, but good design has made it very compact.

The ignition shaft, running at 880 r.p.m., normal, the fan and oil pump shaft running at 1830 r.p.m., and the feed water pump shaft at 435 r.p.m. are coupled together and to the motor by belts. The shafts are adjustable in relation to each other, and are each mounted on a pair of Timken tapered roller bearings to ensure long and trouble-free life under all conditions.

In operation the unit is simple but effective. Fuel oil pumped to the atomiser is by-passed by a fuel control valve until water pressure has been built up by the water pump. When the fuel pressure reaches 60 lbs./sq. inch, the atomise valve opens and the vaporised fuel is ignited by a sparking plug connected to the magneto. The unit is fully equipped with safety devices, and if the steam pressure rises above a predetermined figure, a pressure switch stops the driving motor, restarting it when pressure drops to normal. Similarly excessive superheat causes a by-pass to reduce the full pressure and amount of oil being fed to the burner. Completely automatic control equipment is available.

When it is desired to use the steam generator as a source of hot liquid for cleaning purposes, a second duplex water feed pump with a separate set of control valves is brought into operation. As the fuel feed is constant for any speed, and the supply of water is greatly increased, the boiler is unable to convert it to steam, and produces a hot liquid jet which can be regulated in temperature by control of the suction valve in the second feed water pump.

The boiler will not only handle water, but also mixtures of oil and water, or certain chemical solutions in water. It will thus be seen that this steam generator has many applications quite outside the sphere of the normal steam unit.

The makers are GRESHAM and CRAVEN LIMITED, of Salford, Manchester.

LESS TIN FOR BEARINGS.

SINCE the tin-producing areas in the Pacific have come under Japanese control, the Allies have lost about two-thirds of their supplies and rapid steps are being taken, both in England and America, to reduce consumption. Measured in tons the tinplate industry offers the biggest field for economy, and the manufacture and use of tinplate is being drastically cut by orders restricting the purposes for which tinplate will in future be made available.

The consumption of tin in bearings although much less than in tinplate, appears to offer scope for an even greater proportional saving, but due to the many technical points involved, this saving can only be realised rapidly if users and designers voluntarily co-operate to make the necessary changes in their practice. To this

end the Ministry of Supply had the advice of a small Committee of technical men representing a variety of bearing manufacturers, and after consultation with important classes of users, a report has been drawn up which classifies the various uses of white metal bearings, and recommends for each the most economical tin content that can safely be used in the light of the best present-day practice. An examination of current usage reveals that in many cases alloys are used which are richer in tin than the actual working conditions call for.

A further and very immediate saving can be effected by reducing the thickness of white metal linings. This trend has been followed for some years in automobile design, and it is found that thinner linings carry heavier loads with an increased factor of safety. In few cases is the full thickness of a white metal lining required to allow for wear—nor is thickness in itself much help in the event of failure in lubrication. It is common experience that so long as lubrication is maintained, wear on white metal is almost negligible, and if lubrication breaks down the white metal and almost any quality of backing bronze is cut through rapidly.

The most important point brought out by this report is that it can be given immediate effect to only by the voluntary efforts of bearing users, and the detailed recommendations made will be found very helpful to the designers on whose individual efforts the economy so urgently required now depends. Copies of the report may be obtained from the Joint Controllers of Non-Ferrous Metals, Grand Hotel, Rugby, and from the Tin Research Institute, Fraser Road, Greenford, Middlesex.

CATALOGUES RECEIVED

Miniature Pyrometer Outfits for Small Furnaces.—ELLIOTT BROTHERS (LONDON) LTD., Century Works, Lewisham, London, S.E.13, have issued a 4-page leaflet dealing with miniature Pyrometer outfits which are particularly applicable at the present time in connection with many kinds of muffle furnaces, high speed steel furnaces, and for the heat treatment of tools. All the outfits consist of a miniature temperature indicator of the horizontal edgewise type, with a scale 3½ inches long, and an anti-parallax mirror, together with the appropriate pyrometer stem and leads, according to the maximum temperature to be measured, viz., 1000° C., 1100° C., or 1400° C.

Free copy of the leaflet obtainable upon request.

The Ace Optigage Optical Comparator is an instrument for indicating differences of length by comparison with a master by a system combining mechanical and optical amplification. Differences are indicated by the movement of a line on a screen immediately in front of the operator, in conjunction with a scale. Two adjustable shutters are provided for limiting the scale to the plus and minus tolerances required. Thus, when the indicating line does not appear in the space between the shutters, the work is rejected, or set aside for further examination.

There are two models available, model B for gauge work, and model B1 for parts for which the tolerance is ±a few thousandths of an inch. Both models are provided also with millimetre scale.

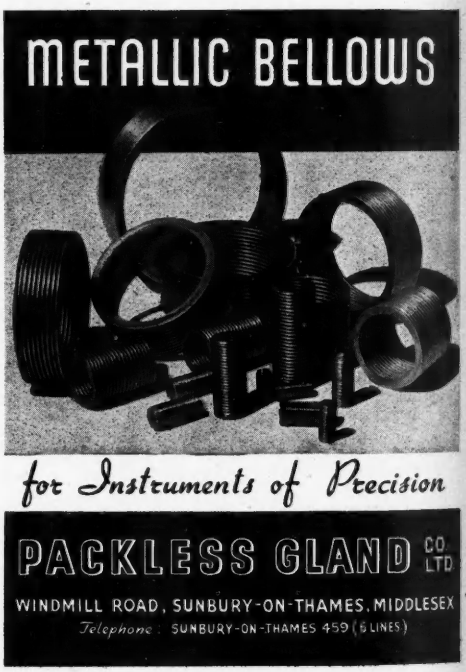
For detailed description of both models of the Ace Optigage Optical Comparator apply to WILLIAM URQUHART, 1023 and 1025, Garratt Lane, London, S.W.7.

New Signalling Controller Tests Condensate Automatically.—At exceptionally low cost, the industrial steam plant can now have dependable protection against damage caused by contaminated condensate, by using Leeds & Northrup's new signalling controller.

Used where indicating and recording of condensate purity are not needed, this simple, automatic equipment tests condensate purity continuously. Its self-contained signal lights show whether condensate is above a specified minimum purity and is safe to use again, or whether it is below this limit and should be diverted to waste.

In addition, the instrument can operate external warning bells or lights. Or, it can provide automatic two-position control by regulating a motor-driven valve in the condensate line to dump impure condensate.

If the operator wishes to measure condensate resistance, he can do so by turning the instrument's control-setting dial until both signal lights are out, and reading the dial scale.

An 8-page catalogue describing this equipment has just been issued. To receive a copy, ask LEEDS & NORTHRUP COMPANY, 4934, Stenton Avenue, Philadelphia, Pa., for Catalogue N-95-163(1), "Simplified Signalling Controller for Automatic Testing of Condensate Purity."

Future Developments in Communication Technique, a paper by P. P. ECKERSLEY, M.I.E.E., will be read at the April, 1942, Members' Meeting of The British Institution of Radio Engineers on Saturday, 18th April, 1942, 3 p.m., at the Federation of British Industries, 21, Tothill Street, London, S.W.1